# THE ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOLUME X.

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REVIEW OF ENGINEERING PROGRESS ABROAD

VOL. X. No. 1 JANUARY, 1949

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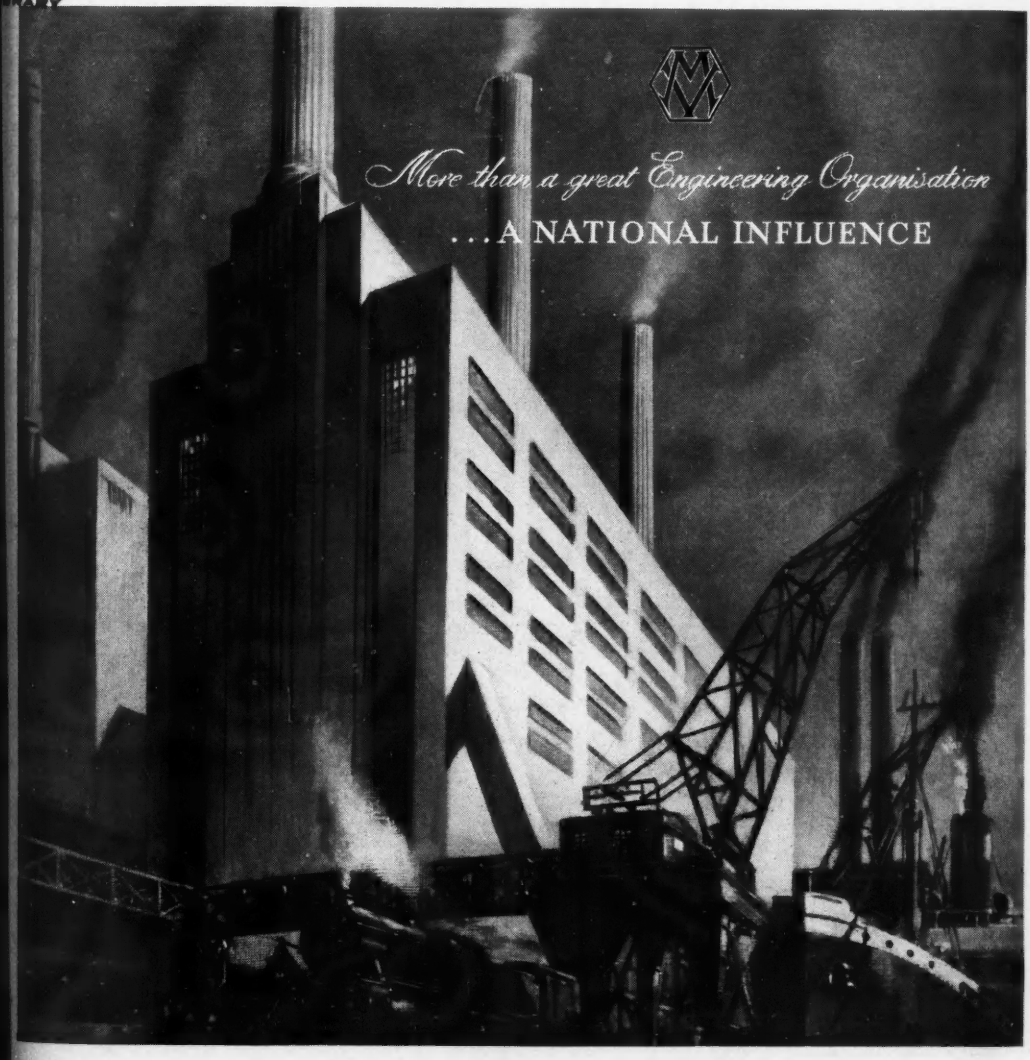


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# IN THIS ISSUE

MOST of the engineering metals in common use are not inherently ductile or inherently brittle, but may change from the ductile to the brittle state, or vice versa, under specified conditions, e.g., if the temperature undergoes a wide change. The Izod notch impact test is the usual test for determining ductility or brittleness under conditions indicated in the corresponding standard specifications. If conditions are altered, e.g., if the temperature at which the test is performed is lowered, the energy absorbed on breaking a ductile material usually drops slowly at first, then passes through a critical phase of widely varying values and, finally, assumes the low values characteristic of a brittle material. The fracture of specimens broken in the critical phase shows a partly brittle and partly ductile character.

This transition from the ductile to the brittle state, which, in the case of ferritic steels, occurs in the region of temperatures experienced under Arctic conditions, received a great deal of attention during the late War. Investigations have been made in the U.S.A. in connection with the fate of welded Liberty ships in Northern waters. The Russian article published on page 7 constitutes an eminent contribution to the study of this problem. The author has conducted static tensile tests at temperatures down to — 196° C. He has found that the same phenomena of transition from ductility to brittleness can be observed in static tests, and has established a certain coherence in phenomena which have hitherto been regarded as rather incoherent. The whole transition range has been investigated and analysed more closely than is possible in impact tests. The theories advanced by the author may not sound convincing to a public used to the more advanced and more penetrating theory of dislocations, but the results of his tests merit consideration not only for theoretical investigations, but also, and even more so, in the practical field of materials testing. The lack of precision in data derived from notch impact tests has long been recognised as a serious drawback to these tests. The tests described here may well offer an alternative for more precise investigation of transition phenomena in engineering metals.

* * *

Prof. Föppl's article on page 11 gives a graphical-analytical method of determining the natural bending frequencies of overhung shafts, and shafts supported by end bearings and carrying one or several discs. The effects of weightless moments of inertia, and of masses with zero moments of inertia, are considered separately, and by addition or subtraction of the values obtained for the natural frequencies for these separate cases, the frequencies of the first, second, third and fourth order vibrations of the complete systems with inertias and mass effects are determined.

* * *

Improper die setting in punch press operation drastically reduces die life, and can be the cause of out-of-parallel press frames, misalignment of ram ways to the press bed and extremely high production and maintenance costs. Some "Do's and Don'ts" for the die setter are given in the article on page 21 and a programme for training die setters is outlined.

* * *

The article overleaf brings a survey of the elements of photo-electric devices which specialists in electronics would consider to be common knowledge. Reasons for including such an article in our journal are, firstly, the ever-increasing scope of this branch of electronics and, secondly, the fact that the importance and the possibilities of photo-electric control equipment are not yet fully realised by practical engineers. Works managers and planning and production engineers can no longer afford to be entirely ignorant of the basic principles of electronics, a knowledge of which is necessary to enable them to recognise the extent to which photo-electric equipment can contribute towards the solution of particular production problems.

The number of man-hours required to produce a particular part largely depends on the extent to which a process can be made automatic. It is claimed that the average productivity of American labour, about two-and-a-half times as high as our own, is due mainly to a more general use of automatic equipment, some consisting of photo-electric devices. The importance which Americans attach to this kind of control equipment is well illustrated by the number of periodicals devoted exclusively to electronics as applied to industry. However, in the last year or two, British electronic firms have made great strides and achieved notable results. To mention only a few, one could name the "Servodyne," a cloth guiding device and control equipment for accurate guiding on cloth processing machinery; a photo-electric tracing unit fitted to the pantograph of a flame-cutter; the application of an infra-red image converter to pyrometry and processing of panchromatic film; various photo-electric cells and switches used for the automatic production of radio sets; and a photo-electric copying device for lathes. The number of possible applications is too wide to be listed here, and the interested reader would be well advised to refer to text books on the subject, as there can be little doubt that photo-electric control equipment will play an even greater rôle in the future.

# Modern Photo-Electric Controls

By J. A. STIEGER. (From *Schweizerische Bauzeitung*, Vol. 66, No. 15, April 10, 1948, pp. 204-208, 18 illustrations.)

PHOTO-ELECTRIC controls are based on the phenomenon by which light reacting on a light-sensitive device produces a change in current flowing through an electric circuit. In particular, two distinctly different types of cells are used in industrial applications, the rectifier type and the emission type of photo-cells.

## 1. THE RECTIFIER CELL

The rectifier, or self-generating, cell in use at the present day consists of an iron disk *a* of about 1 mm thickness, to which a thin layer of selenium *b* is applied (Fig. 1). By a special type of heat treatment, this layer becomes a light-sensitive modification of selenium. The selenium surface is covered with a very thin, transparent, precious metal coating *c*. A metal ring *d* serves as a terminal to the transparent front electrode. Between the selenium surface and the transparent electrode there is an arrestor layer.

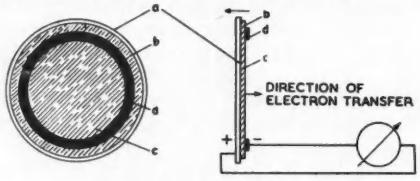


Fig. 1. **Selenium Cell.**
(a) Iron disk ; (b) selenium layer ; (c) skin of precious metal, acting as front electrode ; (d) terminal ring.

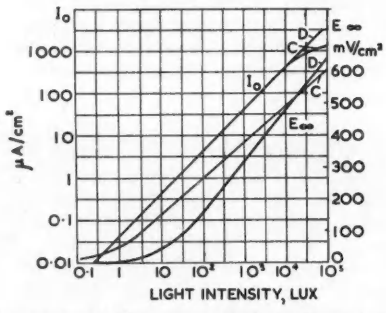


Fig. 2. **No-load voltage $E_\infty$ and short circuit current $I_0$ of a rectifier cell of 1 cm² area.**
Curves *C* relate to standard cell, curves *D* to a special purpose cell.

If a direct current is applied to a cell of this kind so that the iron disk becomes the negative pole and the front electrode the positive pole, a very weak current passes through the selenium layer from the positive to the negative pole. On reversing the polarity, a very strong current flow results. Thus, by applying a voltage from an extraneous source, electrons can pass from the front electrode to the rear electrode, but not vice versa. In Fig. 1 an arrow indicates the direction of their passage.

This behaviour of the selenium cell is utilized for rectifying alternating currents (dry rectifier).

Light penetrating the electrode *c* to the selenium layer produces a photo-electric effect, namely, the release of electrons through the barrier layer to the front electrode *c* (Fig. 1), resulting in the production of an e.m.f.

If an external circuit is connected to the photo-cell, part of the electrons released by the light returns through this circuit to the selenium layer. For a given light intensity, this photo-current is greater, the smaller the resistance of the external circuit. The

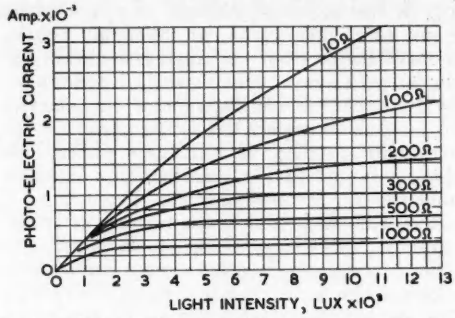


Fig. 3. **Photo-electric current of a selenium cell of 11·5 cm² area as a function of resistance.**

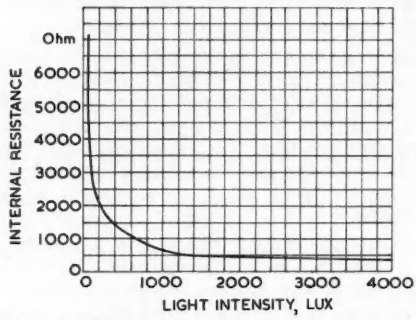


Fig. 4. **Internal resistance of a selenium cell of 11·5 cm² area plotted against light intensity.**

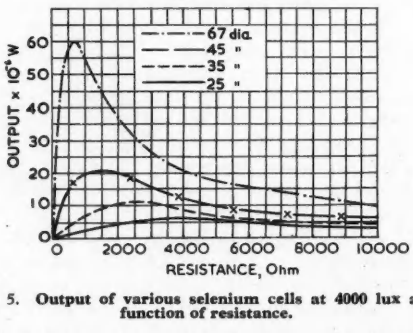


Fig. 5. **Output of various selenium cells at 4000 lux as a function of resistance.**

no-load voltage as a function of light intensity is shown in Fig. 2. With increasing resistance in a closed circuit, the photo-electric current decreases in accordance with Ohm's law, Fig. 3. A graph of the internal resistance as a function of light intensity is illustrated in Fig. 4, while Fig. 5 compares the outputs of various rectifier cells as a function of the external resistance.

The selenium cell is especially well suited for photometric purposes, as it is sensitive over the entire range of visible light rays, Fig. 6. Sensitivity is also good, both in the infra-red and the ultra-violet range.

For the photometric comparison of two light sources, the double photometer is used, Fig. 7, consisting of two rectifier cells joined back to back and exposed to the light-sources *A* and *B* under comparison. The measuring instrument is connected in the circuit of the outer electrodes as a null instrument. Its pointer

should be at zero when both light sources are of the same intensity. Any inequality between the two cells is compensated by adjustable resistances. Very similar in principle is the differential photometer, Fig. 8, consisting of two cells mounted on the same base-plate. The null method is again applied to indicate equality in illumination. This type of photometer is suitable for measuring small mechanical movements.

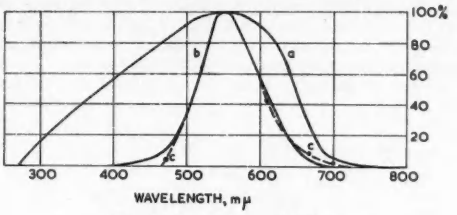


Fig. 6. **Spectral sensitivities.**
(a) Selenium cell; (b) human eye; (c) selenium cell with shield. All curves refer to spectra of constant energy.

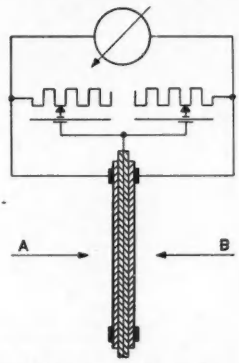


Fig. 7. **Double photo-cell comprising two light sources A and B.**

## 2. THE EMISSION TYPE PHOTO-CELL

The emission type of photo-electric cell consists essentially of two electrodes in an evacuated or gas-filled glass bulb. The shield cathode, coated with a light-sensitive surface, emits electrons. When light falls upon the cathode, the internal resistance of the cell is decreased. Therefore, if a voltage is applied to the cell, the cathode-to-anode transfer of electrons constituting the current follows Ohm's law.

The sensitivity of the cell is defined as the quotient of the electric current in microamperes ($10^{-6}$ A) and the incident energy flow in microwatts ($10^{-6}$ W). The energy flow embraces both visible radiation (light) and invisible radiation (infra-red and ultra-violet). Sensitivity sometimes refers only to the visible part of radiation. The light sensitivity as it is then termed, is given in microamperes per lumen of light radiation.

The sensitivity of a photo-cell is also dependent on the wave length or frequency (colour) of the radiant energy. Fig. 9 shows a typical plot for the sensitivity of two different cells as a function of spectral distribution.

Generally speaking, the sensitivity of a photo-cell can be increased by filling the bulb with a small, precisely dispensed quantity of an inert gas, usually argon. On the other hand, the sensitivity of vacuum cells becomes more uniform with age, and such cells, therefore, are most suited for light-actuated relays and photometric purposes. However, the total electric current in gas-filled cells is a multiple of that in vacuum cells under otherwise equal conditions.

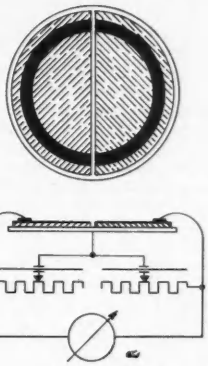

Fig. 8. **Differential photo-cell.**

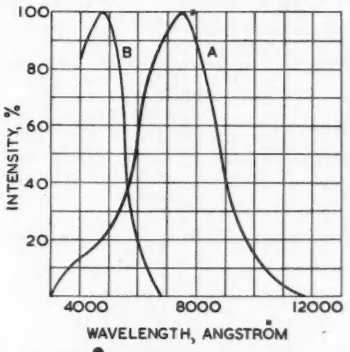


Fig. 9. **Spectral distribution of sensitivities for different types of emission photo-cells.**
*A* with maximum in the red and infra-red range. *B* with maximum in the blue range.

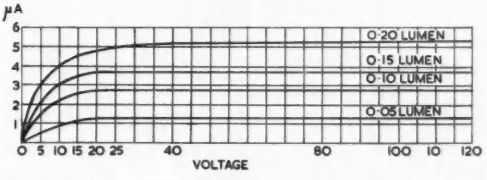


Fig. 10. **Photo-current of vacuum cells as a function of the applied voltage for varying light radiation.**

Characteristic curves for the vacuum cell are shown in Fig. 10, where the photo-electric current is largely independent of the applied voltage and increases in linear relation to light radiation. A strong contrast to these properties can be seen in Fig. 11, showing the characteristic for gas-filled photo-tube. The foregoing considerations apply in the case where the radiation energy flow is of constant magnitude. This, however, might consist of vibrations, e.g., in the case of sound transmission, when the so-called dynamic sensitivity is defined as the ratio of the amplitude of the responding cell current to the incident energy flow. Fig. 13 relates to a sound film unit with photo-electric transmission and, therefore, the sensitivity is given in decibels.

## 3. APPLICATIONS TO PHOTO-ELECTRIC CONTROLS

Photo-electric control can be established by changes in light intensity or interruption in a system consisting

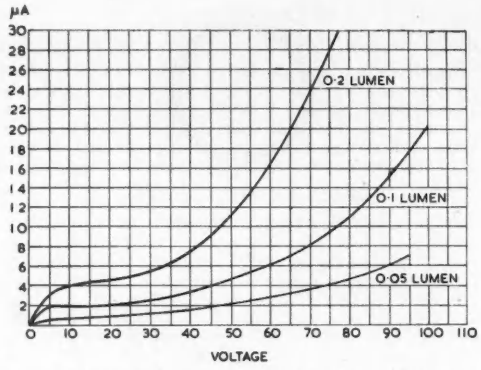


Fig. 11. **Photo-electric current of a gas-filled tube as a function of the applied voltage for varying light radiation.**

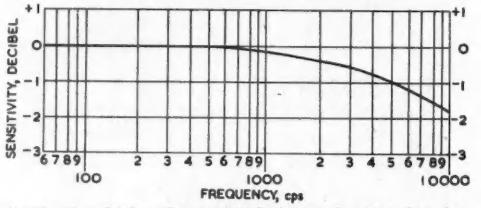


Fig. 12. **Sensitivity of a gas-filled photo-tube plotted against frequency (sound pitch) for a periodically changing light radiation.**

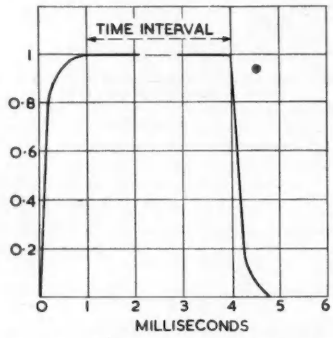


Fig. 13. **Increase and decrease of the relative sensitivity o a gas-filled tube plotted against time.**

of a light source as sender and a photocell as receiver, Fig. 14a, or in a system according to Fig. 14b, where sender and receiver are conveniently combined in one unit and the light ray is reflected back to the unit.

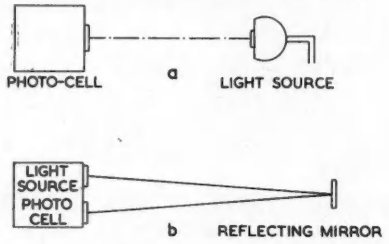


Fig. 14. **Alternative arrangements for photo-electric controls.**

An electric circuit employing an amplifier valve and containing all the elements required for its application to a wide variety of uses is shown in principle in Fig. 15. Relay 4 controls the circuit 5 by the variation in light radiating from the light source 12 to the photocell 13, which is connected to the resistances 7, 8 and 9, the resistance 8 being provided with a tapping. The amplifier tube 6 is energized by the mains through resistance 10, and through the coils of relay 4 connected in parallel with condenser 14. This circuit is controlled by the variable grid potential at point 15, which is dependent of the photo-electric current.

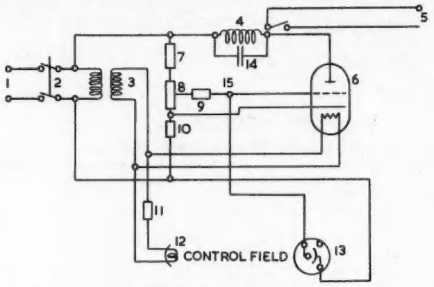


Fig. 15. **Photo-electric control with amplifier tube.**
(1) mains (110V, 50 cycles/sec.)
(2) double-pole mains switch.
(3) transformer.
(4) relay.
(5) power circuit.
(6) amplifier tube.
(7-11) resistances (7 = 8200 Ohm.
8 = 5000 Ohm.
9 = 10 megohm.
10 = 3300 Ohm.
11 according to light source output.)
(12) light source.
(13) photo cell.
(14) condenser (5 to 8 μF).
(15) pin for filament.

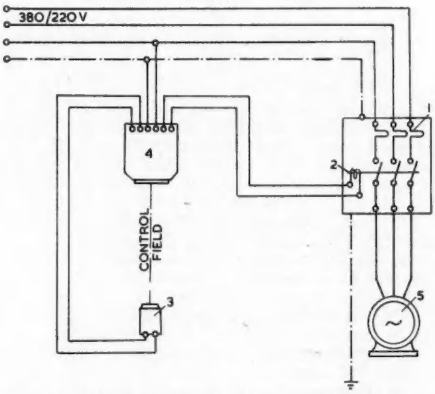


Fig. 16. **Photo-electrically controlled motor.**
(1) Thermal protection; (2) magnetic switch; (3) light source; (4) photo-cell with amplifier tube, transformer and relay; (5) motor.

To illustrate one of the numerous applications, Fig. 16 shows the control of an electric motor 5, where the coil 2 of the motor switch actuates the motor if an object, e.g., a vehicle, enters the control field, the motor being switched in, e.g., to open the gates. The elements shown in Fig. 15 are, with the exception of 13, contained in the photo-cell 4.

The development of photo-electric controls, which has been considerably advanced by war-time research and application, is increasingly being applied to industrial uses.

# Axial Fan with Variable-Pitch Blading

By P. S. DENISENKO. (From *Kotloturbostroyenie*, No. 3, May/June, 1948; pp. 21-23, 5 illustrations.)

THE efficiency of boiler draught fans of the usual centrifugal type is known not to exceed some 50 to 55 per cent; and when operating below the design load, fan efficiency will be still less. In modern central electricity generating stations the power consumption of the draught plant amounts to 35 to 50 per cent of the total power consumption of the auxiliary services, and the need for improved fan efficiency is, therefore, obvious.

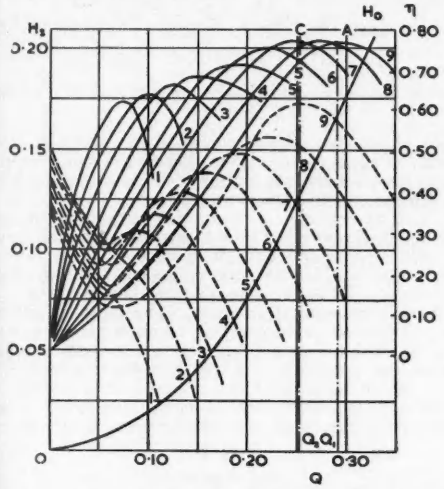


Fig. 1. **Characteristics of axial flow fan with variable-pitch blading (rotor with 8 blades).**
Blades angles :—(1) = 10°; (2) = 15°; (3) = 20°; (4) = 25°; (5) = 30°; (6) = 35°; (7) = 40°; (8) = 45°; (9) = 50°.
Solid lines :—efficiency characteristics.
Broken lines :—delivery head characteristics.

A comparison of the respective characteristics of an axial fan and a centrifugal fan of the Sirocco type shows that the characteristics of the axial fan are considerably steeper than those of the centrifugal machine, although the efficiency of the axial fan at the design load point is considerably higher. Owing to their unfavourable steep characteristics, axial fans are not employed in boiler plant services. Very high operating efficiencies can, however, be maintained with axial fans over a wide load range by equipping them with variable-pitch blading. The characteristics of an axial flow fan with variable-pitch blading are charted in Fig. 1, which is based on a fan of 850 mm diameter running at 900 rpm. Both the delivery head and the efficiency characteristics are charted for blade angles ranging from 10 to 50 degrees, and it will be noted that almost constant fan efficiency can be obtained over a load range from three-quarter to full load by changing the pitch angle from 30 to 50 degrees. At point $A$ of the characteristic curve 9, a delivery $Q_1$ against a head $H_s$ obtains for a blade angle of 50 degrees. A smaller delivery $Q_2$ for the same blade angle will correspond with point $B$ on curve 9, indicating a drop in fan

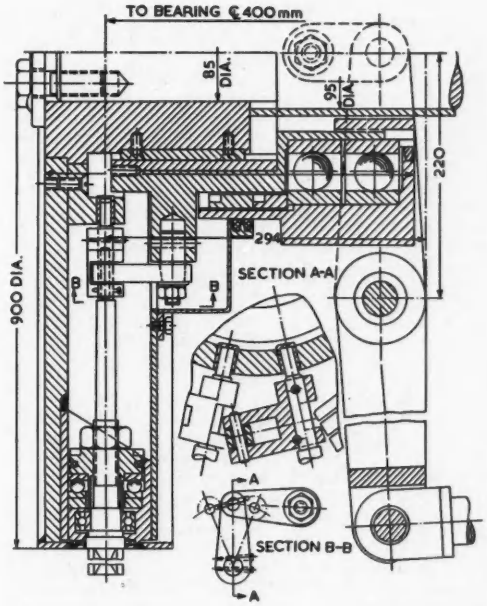


Fig. 3. **Blade adjusting mechanism.**

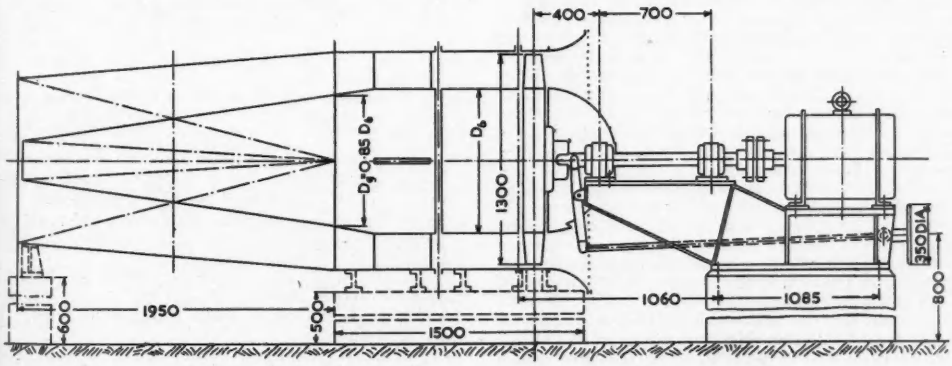


Fig. 2. **Overall arrangement of axial-flow fan with variable-pitch blading.**

efficiency. However, the delivery $Q_2$ can also be obtained with practically unimpaired efficiency by changing the blade angle to 40 degrees. The operating condition will now be expressed by point *C* on curve 7.

A draught fan with variable-pitch blading is under construction and will be installed at the experimental boiler plant of the Leningrad Polytechnic Institute "Kalinin." The design data of this fan are as follows: Delivery, 50,000 m³/hr against 224 mm water gauge. Overload capacity, 10 per cent. The diameter of the rotor over the blade tips is 1300 mm, and there will be 16 blades in all. Design efficiency is 72 per cent, with a fan speed of 1500 rpm. Power consumption is given as 45 kW.

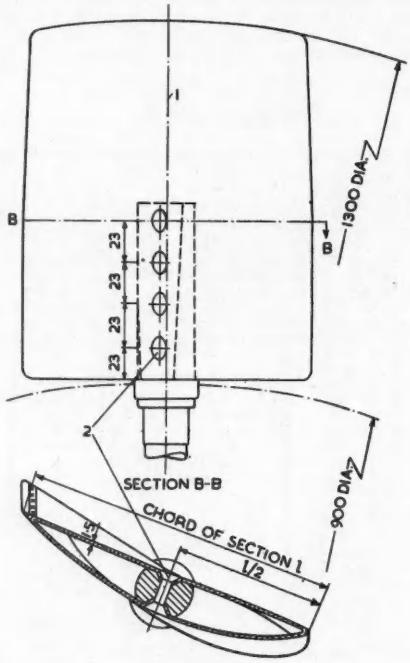


Fig. 4. **Variable-pitch blade.**

Twenty-one stationary guide vanes are placed on the discharge side of the rotor. These vanes are made of mild steel plate of 3 mm thickness. The guide vanes are 205 mm high and, over a length of 200 mm on the inlet side, make an angle of 28° 30′ with the longitudinal axis of the fan. The fan rotor and the guide-vane ring are placed in a cylindrical section of the diffuser. The latter is composed of several flanged sections and has an overall length of 3·05 m. Referring to Fig. 2, it will be noted that a suction nozzle is provided on the inlet side of the fan casing. The importance of the use of such a suction nozzle has been confirmed by tests carried out at the Central Aerodynamic Institute, where it was found that the omission of a suction nozzle results in a reduction in delivery pressure and efficiency by 10 to 15 per cent.

The design details of the variable-pitch mechanism are shown in Fig. 3. Each blade is carried on a shaft of 415 mm overall length, the shaft extending 100 mm into the blade. The hollow blades (Fig. 4) are made of steel sheet of 1·5 mm thickness. The profiles of the guide vanes as well as the fan blades were determined by the method developed by Prof. I. N. Voznesenko of the Hydrodynamic Institute of the Leningrad Polytechnic Institute "Kalinin." In axial fans the clearance between the blade tips and the casing is of considerable influence upon the delivery pressure, large clearances causing a considerable drop in delivery pressure. At the same time, the clearance must be made the smaller, the larger the peripheral speed of the blade tip. In the design referred to, a clearance of 2·5 mm is adopted.

## New Design for Steam Separation in High-Pressure Boilers

(From *Kotloturbostroyenie, Russia*, No. 2, 1948, p. 32, 1 illustration.)

THIS design, recently developed in the U.S.S.R., has been installed in one 60-at installation and several 32-at installations, and has been found to be superior to other drum-type steam separators.

The new design is based on recognition of the paramount importance of the secondary condensation of steam. The water-steam mixture is conducted to the baffles both in the steam space and water space of the drum. It will be seen that the bottom baffle (1), Fig. 1, is mounted at an angle. The large volume of water promotes rapid damping of water currents, and the considerable area of evaporation means that the steam emerging from the gap between the baffle (2) and bottom baffle (1) has a relatively high degree of dryness. The intermediate baffles (3) divide the steam flow, thus improving the effectiveness of the steam

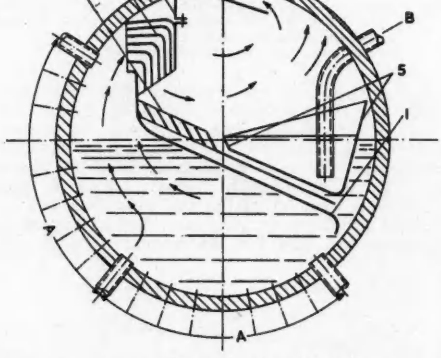


Fig. 1. **Arrangement of baffles and trough for prevention of foam in the boiler drum.**
A—suggested arc for boiler tubes. B—feed-water inlet.
The arrows indicate the steam path.

space beyond the bottom baffle. Drops of water carried by the steam fall from the intermediate baffles on to the louvre (4) at an angle which prevents their further disintegration. The effect of the louvre has been found to be considerable.

A trough (5) is installed to prevent the formation of foam due to the feed-water, and this proves to be effective, as the boiler water mixes only slowly with the feed-water.

The new equipment can be simply installed and is easily accessible for inspection.

# An Investigation of the Stress-Strain Diagram at Low Temperatures

By E. M. SHEVANDIN. (From *Zavodskaya Laboratoria*, Vol. 13, No. 7, 1947, pp. 858-870, 12 illustrations.)

THE true-stress diagram was obtained by the usual method on a Gagarin press, with tensile test specimens of the normal shape. At various times during the tests the specimens were taken from the test jig and carefully measured with an accuracy of 0·01 mm, the diameter being determined in two directions at right angles. The corresponding load was taken from the diagram. The low-temperature tests were made in a special tin vessel of tumbler shape with a thick flat bottom, possessing good thermal insulation properties. The tension shackles with the specimen were kept in this vessel during the tests, cooling being obtained by pouring liquid air into petrol or on cotton wool enveloping the specimen. The tension shackles with the specimen were precooled to 10-20° C below the specified test temperature, and loading began when the system had re-gained the chosen temperatures on warming up. Temperature fluctuations were not appreciable when employing this method, and deviations could be kept within permissible limits.

The materials tested were (1) 0·12 per cent carbon steel, normalized at 920° C, and tempered at 600° C ; (2) ditto, over-annealed at 1200° C (coarse-grained) ; (3) 0·2 per cent carbon steel, annealed at 950° C ; (4) ditto, over-annealed at 1200° C (coarse-grained) ; (5) 0·4 per cent carbon steel, normalized at 870° C, tempered at 600° C ; (6) ditto, over-annealed at 1050° C (coarse-grained); (7) nickel steel, 5 per cent Ni, untreated; (8) copper annealed at 650° C.

The most striking feature of the diagrams obtained at different temperatures (Fig. 1) is the marked extension of yield at lower temperatures.

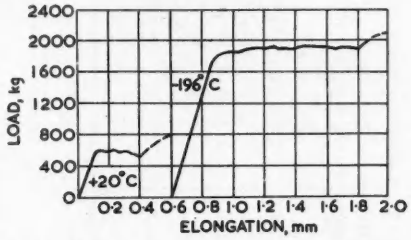


Fig. 1. **Initial part of the conventional stress-strain diagram, taken at + 20 and — 196° C respectively.**

For mild steel (0·12 and 0·2 per cent C) the change of temperature from + 20 to — 196° C increases yield 2-4 fold before work-hardening commences and raises the yield point appreciably. In the interval shown, the yield stress increases 3·5 times, whilst the ultimate strength is only doubled. Nickel-steel and copper behave quite differently, as the yield stress increases only to an insignificant extent and more slowly than the other characteristics shown. Ni-steel shows only a slight plastic yield, copper none at all. Ludwik's explanation[1] of the steep increase of the yield point at low temperature as being due to diminution of solubility and precipitation from the solid solution, resulting in a finely dispersed second phase, is improbable, considering the low degree of mobility of the atoms at low temperatures and the corresponding difficulties in phase formation. Davidenkov's[2] explanation of the phenomenon by greatly increased ferrite hardness and ensuing reduction of stress concentrations, this forcing the yield point up, is acceptable as a contributory factor, but not as a basic one. Al-monocrystals, tested by Yakutovich at varying temperatures, showed the following behaviour: at high temperatures the plastic deformation took place in the form of a small number of widely-spaced steps, but at low temperatures, in the form of a great number of extremely small shifts. On the other hand, the spread of the elastic deformation in its initial stage is impeded by the lattice of tertiary cementite which is deposited along the ferrite grain boundaries according to Köster's theory, for the acceptance of which Davidenkov puts over quite a convincing case. Hence, a combination of Köster's and Yakutovich's theories seems to furnish a perfectly satisfactory and consistent explanation.

The non-uniform character of the deformation and its concentration on individual grains after the elastic limit has been exceeded, but before the yield point has been reached, is well-known. Since the steepness of the initial rise of the stress, with respect to the area reduction, increases with reduced temperature, as shown by the start of the curves in Figs. 2-9, the non-uniformity will be evened out and the rate of yield spread increased; more grains are affected and more spots within the grain participate in the yield. The higher, but localized pressure on the lattice will decrease at a lower temperature. For equal brittle strength of the cementite, this results in a higher load being required for breaking down the lattice, i.e., increased yield point. This increase must fully manifest itself prior to the appearance of the first Lüders line, whereas after appearance of the full line, the process of plastic yield is determined also by stress concentration at the steps formed by the Lüders lines. This is evidenced by Goerens and Mailander's experimental results, which show that with lower temperatures, the upper yield point rises more markedly than the stress at the level portions of the yield. The true-stress area reduction diagrams of the materials investigated are given in Figs. 2-9.

After, and because of, the particularly pronounced *initial* rise of stress with lower temperatures, the stresses in the adjoining part of the diagram (i.e. at relatively small deformations and even before necking commences, see Figs. 2 to 9) increase more slowly at low than at high temperatures. This is again reversed in the final part of the curves, at greater deformations. The curve joining the end points, corresponding to breaking stress ($\sigma_z$) at various temperatures, comprises two branches, one ascending and one descending. The latter can be separated into a straight vertical and a curved part, each of which has a particular aspect of the fracture as its characteristic.

Part *AB* (see theoretical true-stress diagram Fig. 10) is characterized by bright crystalline surfaces of the fracture becoming finer, smoother, and more homogeneous on progressively approaching *B*. This difference is determined by the amount of plastic deformation suffered prior to failure, since the grain size is reduced with increasing deformation. Part *CD* is the locus of purely dull and fibrous fractures. Branch *BC* represents semi-brittle fractures; one part of the cross-section has a crystalline appearance, and the other a dull fibrous appearance. Brittle and viscous zones always follow each other in a definite order, viz., near the rim the brittle zone of fracture, and at the centre the dull eye-shaped zone of fracture. At lowered temperatures, near *B*, the brittle character becomes more and more marked, while the eye-shaped zone contracts more and more until it finally disappears. The phenomenon can be explained by the fact that, once a neck has been formed, the destruction occurs in a three-dimensional stressed state, with the maximum axial stress at the centre. Hence the first crack originates here[4,5]. The crack takes the eye-shaped form

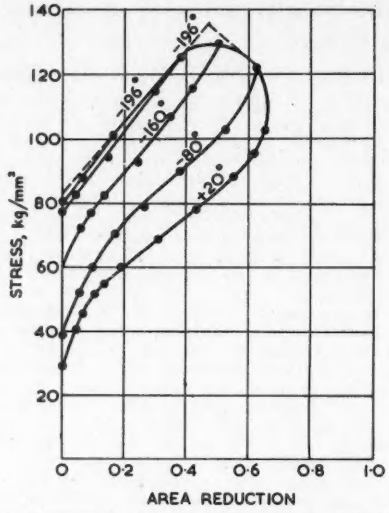


Fig. 2. **Diagram showing true stresses plotted against area reduction at lowered temperatures for steel containing 0·12 per cent C, normalized at 920° C and tempered at 600° C.**

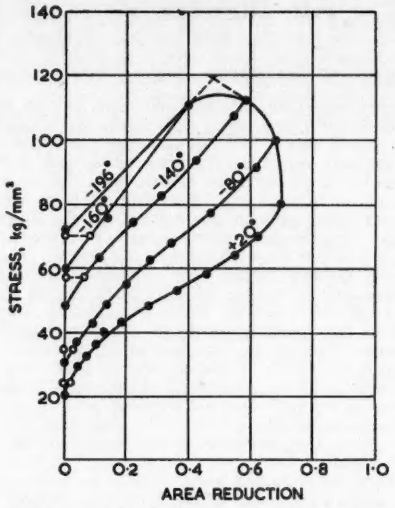


Fig. 3. **Diagram showing true stresses plotted against area reduction for steel containing 0·12 per cent C, over-annealed at 1200° C.**

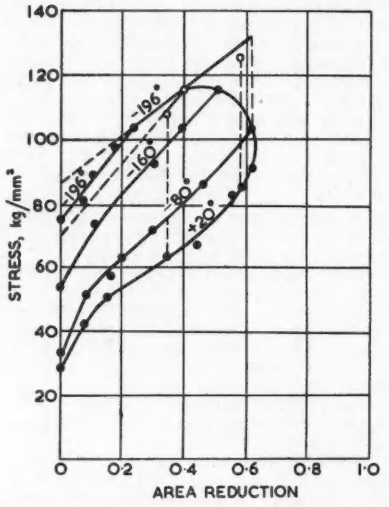


Fig. 4. **Diagram showing true stresses plotted against area reduction for steel containing 0·2 per cent C, annealed at 950° C.**

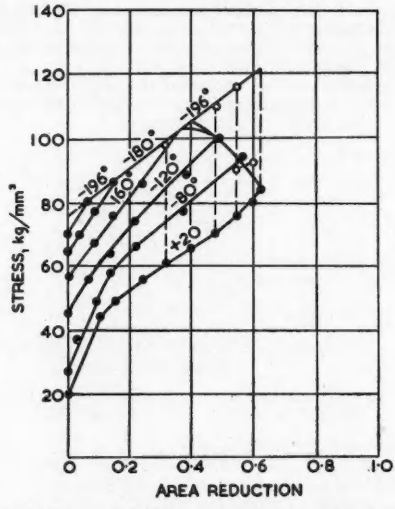


Fig. 5. **Diagram showing true stresses plotted against area reduction for steel containing 0·2 per cent C, over-annealed at 1200° C.**

and plays the part of an original internal notch, with all its characteristics, such as stress concentration and intensified three-dimensional stressed state. This, jointly with the low temperature, raises the stress to the level of the resistance to brittle fracture (continuation of *AB*) and leads to premature, brittle failure. The lower the temperature, the smaller the internal notch required to cause failure. From *C* onwards, this occurs with decreasing contraction, thus foreshortening the true-stress diagram. Imagining this process cut out, the diagram would end on *CE*, the continuation of *CD*, and if the failure took place on *BC*, the meaning of *DE* would be entirely viscous destruction of the metal. All this corresponds exactly to Davidenkov's and Fridman's theory and predictions. Yet, according to our own experiments, it has to be stated that the strength curve has, not two, but three branches, the third of which corresponds to semi-brittle failure and can be explained by the transition from the isolated fibre to which Davidenkov's theory strictly applies, to the whole specimen. It corresponds, as can be shown, to the critical interval of semi-brittle fracture in notch-impact tests.

After plastic deformation at room temperature up to values corresponding to those of interval *BE*, several specimens were tested to destruction at — 196° C, i.e. in conditions prejudicial to plastic deformation. The experimental data, plotted in Figs. 4 and 5, provides indisputable evidence that line *BE* determines the value of the brittle strength as a function of the residual deformations, a result obtained by a different approach by Davidenkov and Sacharov[7]. Other specimens de-

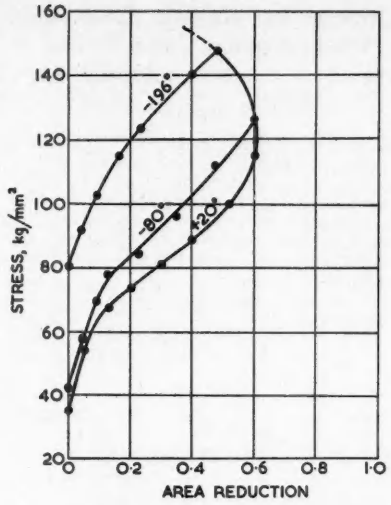


Fig. 6. **Diagram showing true stresses plotted against area eduction for steel containing 0·4 per cent C, normalized at 870° C and tempered at 600° C.**

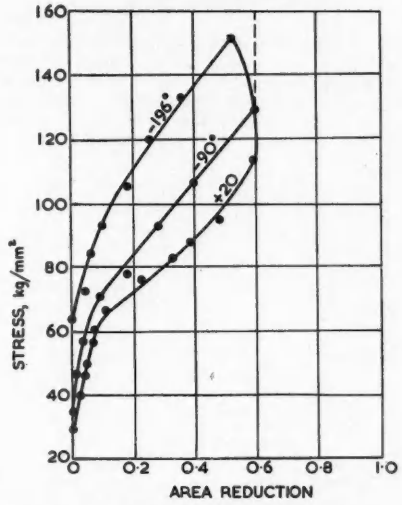


Fig. 8. **Diagram showing true stresses plotted against area reduction for a 5 per cent untreated nickel steel.**

formed at + 20° C to values corresponding to intervals *AB*, then cooled to — 196° C, showed a slight additional plastic deformation at the low temperature and finally failed on the level of line *AB*. The conception of monotonous and steady increase of the resistance to brittle failure with an increasing degree of plastic deformation goes a long way towards explaining the shape of the strength curve. Actually the resistance curve has only one branch, *ABE*, the end point *E* of which has special significance. It corresponds to the stress which (at liquid-air temperature) marks the hypothetical failure of a specimen previously subjected to the deformation corresponding to failure at room temperature. From this point of view, there is actually a discontinuity between the end of the true-stress diagram and the strength curve. *DE* has a physical meaning only for

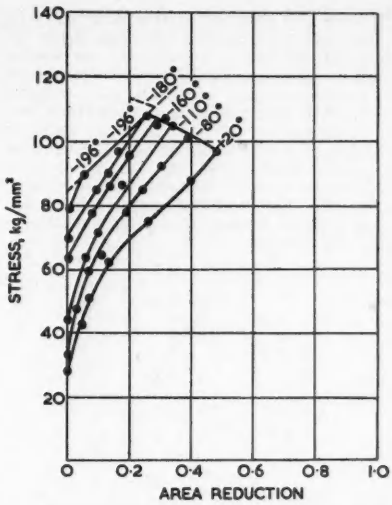


Fig. 7. **Diagram showing true stresses plotted against area reduction for steel containing 0·4 per cent C, over-annealed at 1050° C.**

failure at low temperature. In actual fact, however, it is transformed into curve *BCD*.

The fact that the region corresponding to viscous strength is a vertical straight line proves the law of failure by maximum strain. This is proved by tests on copper specimens which, at every temperature, failed

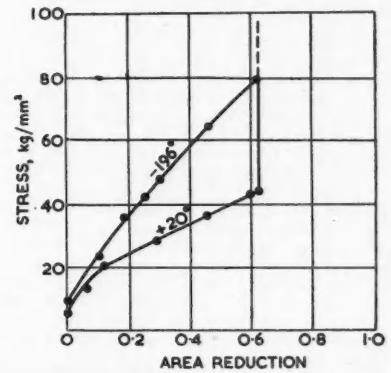


Fig. 9. **Diagram showing true stresses plotted against area reduction for copper annealed at 650° C.**

on line *CD*. The whole range of phenomena may only be observed on materials exhibiting a propensity to brittle failure. For materials of lesser brittleness only *BC* and *CD* are obtainable, but not *AB* (e.g. Ni-steel, Fig. 8). For typically viscous materials such as copper, only *CD* remains (Fig. 9). *CD* again, is missing for cold-short metals (Figs. 5 and 7). For further increasing cold brittleness, *BC* and *AB* will disappear and the whole strength diagram will contract towards the point *A*, corresponding to brittle failure, and its ordinate to the brittle strength of the material in its undeformed condition.

The influence of previous deformation on the tendency to brittleness is shown by the behaviour of a test piece which, when pre-deformed at room temperature and then cooled to — 196° C, showed a slight additional

deformation before failing on the level of line *AB* (Fig. 4). One would have expected the specimen either to fail instantaneously on cooling or to carry stresses higher than those of level *AB*, i.e., to follow a continuation of the original true-stress curve of a test taken entirely at — 196° C. The test result shows, however, that the previous deformation suffered at room temperature cannot have contributed to failure to the same extent as the last part of the deformation, i.e. that suffered at — 196° C. One can say, therefore, that in destructive testing, not the sum of all deformations, but only deformations suffered immediately prior to failure constitute the main danger.

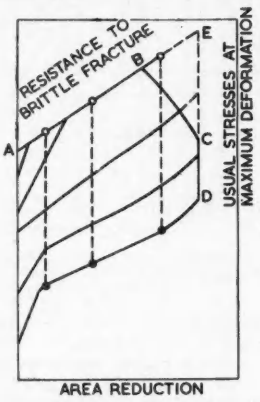


Fig. 10. **Graph showing the transition from the ductile to the brittle state. Diagram of true stresses at lower temperatures and limiting curve of strength.**

The sloping curve of fibrous failure in Davidenkov's representation must be supplanted by a vertical line. This obviates the difficulty in explaining the fall in the resistance to failure for the viscous, i.e., ductile case.

Tendency to brittleness is characterized:—

(a) By the level of line *AB*, Fig. 10, i.e. by the true stress at which a material fails in entirely brittle fracture under different conditions of temperature and plastic deformation suffered during, or previous to, the tensile test. Line *AB* can be called the resistance against brittle fracture over the whole range in which entirely brittle fracture can occur.

(b) By the distance of line *AB* above the true-stress curve at, say, room temperature. The greater this stress difference, the smaller the inclination to brittleness at any given stage of plastic deformation.

(c) The rate of stress increase will decrease with decreasing temperature when a test piece is cooled at a certain stage of *appreciable* plastic deformation during the test. The higher this stress increase for a given temperature interval, the quicker the resistance line *AB* will be reached, and the greater the tendency to cold brittleness.

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# Removal of Heavy Overhead Transmission Line Poles

By V. Gerzselyi. (From *Elektrotechnika, Hungary*, Vol. 40, No. 9, September, 1948, pp. 207-209, 8 illustrations.)

The Budafok Electricity Co. had to remove 50 reinforced concrete poles from their position in a transmission line and re-erect them for another line. The poles were 35 ft long and weighed 39 cwt each. A two-wheeled truck with rubber tyres (shown in Fig. 1, dimensions in mm) was used for lifting out and re-erecting the poles and could also be adapted for their transportation.

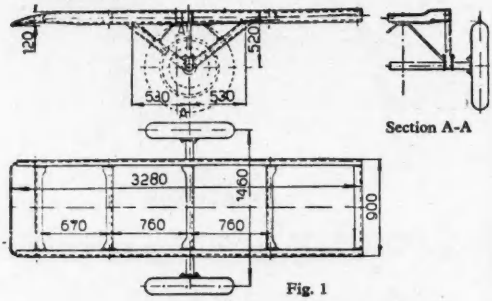



Fig. 1

The method of handling is shown in Fig. 2. The pole is secured to the truck so as to place its centre of gravity as near as possible to the axle of the truck. A wooden stand of corresponding height is set up at the pole pit. To safeguard the poles against breakage during erection, long rigid supporting plates are fixed to them.

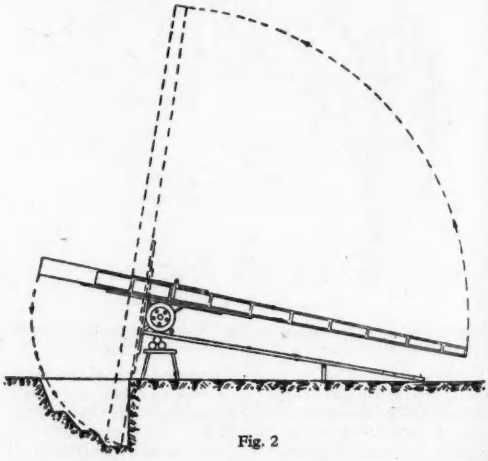


Fig. 2

When the pole has been secured to the truck, the truck is pulled onto the stand. Erection is carried out with the aid of four guy ropes. When the pole is almost vertical, it is detached from the truck and erection is completed in the usual way. The lifting-out procedure is the reverse of that adopted during erection.

It is claimed that the number of workmen required for removal was reduced 25 per cent by using this truck, the speed of the operation being doubled. The risk of breaking poles is also minimized.

# Graphical Determination of Critical Speeds of Multi-Mass Systems

By O. FÖPPL. (From *Die Technik*, Vol. 3, No. 6, June, 1948, pp. 253-256, 9 illustrations.)

## 1. PRELIMINARY REMARKS

THE object of this report is to determine the critical speeds of turbine shafts performing flexural vibrations. Each running speed $n_W$ of the turbine shaft is related to a natural frequency $n_B$ of the flexurally vibrating system. The amplitude of vibration will build up if an external impulse is allowed to act on the system with a frequency equal to $n_B$. It will also build up without external forces if the running speed is equal to the bending frequency $n_B$, as a result of the unbalanced forces due to the inevitable eccentricity $e$ of the shaft in its bearings, or the momentary deviation of the instantaneous axis of rotation from the principal axis of inertia of the shaft. This critical running speed will be denoted by $n_{WK}$, so that $n_{W\,crit} = n_{WK} = n_B$. Furthermore, the following notation will be used: $\Theta_y$ = mass moment of inertia for bending, about the $y$-axis; $\Theta_x$ = mass moment of inertia for torsion, about the $x$-axis; $i_y$ and $i_x$ = corresponding radii of gyration; $n_0$ = critical speed for the limiting value $i = 0$.

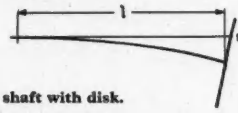



Fig. 1. **Overhung shaft with disk.**

If we consider the overhung shaft with a disk as shown in Fig. 1, we can determine its vibratory characteristics by a method indicated by C. B. Biezeno and R. Grammel in "Technische Dynamik," p. 807, J. Springer, Berlin, 1939. The narrow disk has the moment of inertia $\Theta_y = \Theta_x/2 = m\,i_y^2$, and the bending vibrations of the overhung system are represented by the diagram of Fig. 2. For any such shaft the critical ratio $n_{WK}/n_0$ can be determined graphically from these curves after finding the ratio $\xi = i/l$ of the system. The effect of $\xi = i/l$ on $n_{WK}$ is indicated for in-phase and opposite-phase vibrations of the 1st and 2nd modes, and it will be noted that only in-phase vibrations are dangerous at critical speeds, as self-excitation can only be caused by vibrations of this type. The practical

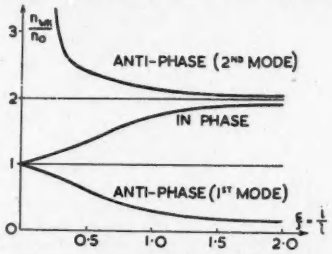



Fig. 2. **Frequency ratio $n_{WK}/n_0$ of vibrating system shown in Fig. 1, according to Biezeno and Grammel.**

value of Fig. 2 is restricted, however, by the fact that, in practice, several other parameters besides $\xi = i/l$ will have to be taken into account for disks of a certain width and for non-overhung systems, so that several diagrams would be necessary.

## 2. THE FREQUENCY DIAGRAM

Considerably more information is obtainable from the frequency diagram of Fig. 3, which shows the relation between $n_W$, or $n_B$, and the reduced radius of gyration $i_{red}$ or the non-dimensional quantity $z = \Theta_{red}/(ml^2)$ In this diagram, $i_{red}$ represents the radius of gyration which would be allotted to the disk on the non-rotating shaft in order to obtain the same vibrations as on the rotating shaft.

The natural frequency $n_1$ of the shaft with several concentrated masses is determined in the usual manner, and then for each running speed $n_W$ or each ratio $n_W/n_1$ we refer to Fig. 3, which is only valid for the special case $i_y = l$, in order to obtain the value $i_{red}$ corresponding to a determined bending frequency $n_B$. In the system considered in Fig. 3, for instance, the critical speed $n_{WK_1}$ is situated at the point of intersection of the two curves $n_W = f_1\,(i_{red})$ and $n_B = f_2\,(i_{red})$, over the abscissa $z = -1{\cdot}0$.

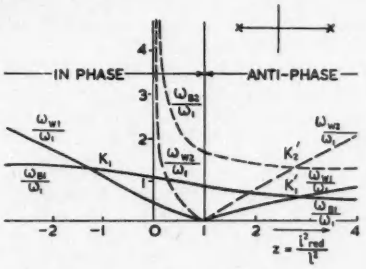



Fig. 3. **Frequency diagram of a shaft with disk between the bearings.**

The frequency diagram of Fig. 3 shows, in particular, that two intersections of the $n_B$ and $n_W$– curves situated one above the other would be obtained at a point $\Theta_{red} > 0$ or $z > 0$ (that is, two critical speeds $n_{WK_1}$ and $n_{WK_2}$), if $\Theta_x - \Theta_y > 0$. However, in the special case covered by Fig. 3 (narrow disk for which $\Theta_y < \Theta_x$), the curves of the 2nd order $\omega_W = f_W\,(z)$ and $\omega_B = f_B\,(z)$ do not intersect; but it is immediately apparent that when $\Theta_y > \Theta_x$ the 2nd order curves will also intersect. For a wide disk, the conditions are, therefore, different from those obtaining for narrow disks which are based on the implicit assumption $\Theta_y < \Theta_x$. Each individual mass situated on the shaft can, in fact be related to one critical speed $n_{WK_1}$ lower than $n_1$ (natural frequency when $i = 0$), and a critical speed $n_{WK_2}$ which is higher than $n_1$ and also higher than $n_2$ ($n_2$ = natural frequency for which $m = \Theta/i^2 = 0$). The following conclusion can, therefore, be drawn: Whereas in the Biezeno-Grammel diagram (Fig. 2), the moments of inertia are varied, in the frequency diagram of Fig. 3 equivalent moments of inertia $\Theta_{red}$ are introduced, which have a magnitude depending on the running speed $n_W$ of the shaft. As a result of the variability of the moments of inertia, the non-rotating shaft will bend under natural or free vibration conditions in the same way as the same shaft with the given true moments of inertia $\Theta_x$ and $\Theta_y$ rotating at various speeds $n_W$. The frequency diagram, constructed only for a given vibrating system with various running speeds, makes it possible to derive considerable information as to the behaviour of the rotating shaft at various speeds. The detailed calculation of this diagram is given in Technical Note No. 40, Part 1, issued by the Wöhler Institute, and also in Schweizerische Bauzeitung, No. 3, March, 1948, Vol. 66, p. 31.

## 3. SHAFT WITH SEVERAL MASSES

Similar considerations apply to the case of a shaft with several masses. All the conclusions previously made are valid, provided that for each individual mass $\Theta_y > \Theta_x$. A frequency diagram can also be con-

structed in this case. As an example, we shall consider a shaft supported by two bearings (see Fig. 4). In the vibration of the non-rotating shaft, the forces $P_1$ and $P_2$ and the moments $M_1$ and $M_2$ act at points 1 and 2. The deflection $\xi_1$ at point 1 consists of an angular deviation $\varphi_1$ of disk 1, due to deviations $\varphi_{11}$ and $\varphi_{12}$ caused by the forces $P_1$ and $P_2$, and the deviations $\varphi_{13}$ and $\varphi_{14}$ due to the moments $M_1$ and $M_2$.

We have, therefore,

$$\left.\begin{aligned} \xi_1 &= \alpha_{11} P_1 + \alpha_{12} P_2 + (\alpha_{13}/l) M_1 + (\alpha_{14}/l) M_2 \\ \xi_2 &= \alpha_{21} P_1 + \alpha_{22} P_2 + (\alpha_{23}/l) M_1 + (\alpha_{24}/l) M_2 \\ l\varphi_1 &= \alpha_{31} P_1 + \alpha_{32} P_2 + (\alpha_{33}/l) M_1 + (\alpha_{34}/l) M_2 \\ l\varphi_2 &= \alpha_{41} P_1 + \alpha_{42} P_2 + (\alpha_{43}/l) M_1 + (\alpha_{44}/l) M_2 \end{aligned}\right\} (1)$$

The influence coefficients $\alpha_{jk}$ have the dimension cm/kg, and a certain number of these coefficients will have the same value for a given system.

The natural frequencies $n_B$ of the entire system are attained when the masses $m_1$ and $m_2$ oscillate with the same angular velocity, that is, when

$$\left.\begin{aligned} \omega^2 = P_1/(\xi_1 m_1) &= P_2/(\xi_2 m_2) = M_1 l/(l\varphi_1 \Theta_1) \\ &= M_2 l/(l\varphi_2 \Theta_2) \end{aligned}\right\} (2)$$

In dealing with a vibration of a given order (for instance, a 1st order vibration), one quantity, for instance $P_1$, can be chosen arbitrarily since in this calculation only the angular velocity $\omega$, and not the deflections $\xi$ or $\varphi$, requires determination. The other three quantities must be chosen in such a way that they will satisfy the three equations (2), and in this manner, three coefficients $K$ are determined

$$K_2 = P_2/P_1,\ K_3 = M_1/(P_1 l),\ K_4 = M_2/(P_1 l) \quad (3)$$

We obtain three equations when we substitute the values of eq. (3) into eq. (2), taking eq. (1) into account, and these equations give four natural frequencies for the non-rotating shaft, which are all real if the masses $m_1$ and $m_2$, and the moments of inertia $\Theta_1$ and $\Theta_2$, have positive values. Thus we have determined four points of the frequency diagram $\omega_B = f(\Theta_{red_1})$, which, for instance, can be related to a non-rotating shaft ($\omega_W = 0$) for which $\Theta_{red_1} = \Theta_1$ and $\Theta_{red_2} = \Theta_2$, so that $\Theta_{red_2} = \gamma\, \Theta_{red_1}$. The ratio $\gamma$ is taken into account in the frequency diagram by the fact that the values of $\Theta_{red_2}$ and $\Theta_{red_1}$ coincide on the axis of the abscissae.

When the shaft rotates with an angular velocity $\omega_W$, the true moments of inertia $\Theta_{y1}$ and $\Theta_{y2}$ are replaced by the equivalent moments of inertia $\Theta_{red_1}$ and $\Theta_{red_2}$ in accordance with the relations

$$\omega_0^2\, \Theta_{red} = \Theta_y\, \omega_B^2 - \Theta_x\, \omega_B\, \omega_W = \omega_B^2\, \Theta_{red}, \quad .. \quad (4)$$

where $\omega_0$ is the natural angular velocity of vibration of the non-rotating shaft, and $\omega_B$ the natural angular velocity of the rotating shaft. In eq. (4) we have put $\omega_0 = \omega_B$, and the moments of inertia $\Theta_{red}$ are thus determined in such a way that, when $\Theta_y$ is replaced by $\Theta_{red}$, we have the equation $\omega_0 = \omega_B$. The reduced moments of inertia $\Theta_{red_1}$ and $\Theta_{red_2}$ can now be determined from eq. (4):

$$\Theta_{red_1} = \Theta_{y1} - (\omega_W/\omega_B)\, \Theta_{x1} \quad .. \quad .. \quad (5a)$$

$$\Theta_{red_2} = \Theta_{y2} - (\omega_W/\omega_B)\, \Theta_{x2} \quad .. \quad .. \quad (5b)$$

In Fig. 5, $\omega_{B1}$ ($= 2\pi/60\ n_{B1}$) is plotted as a function of $\Theta_{red_1}$. The $\omega_B$ curves also include the curves giving the variation of $\omega_W/\omega_B$ according to eq. (5a) as a function of $\Theta_{red_1}$ or of $i^2_{red}/l^2$. The relation between $\Theta_{red_1}$ and $\Theta_{red_2}$ when $\Theta_{red_1}$ is varied is obtained from eq. (5b). The values of $\omega_B$ are replaced in this figure by the non-dimensional ratio $\omega_B/\omega_1$.

Fig. 4. **Shaft with two disks between the bearings.**

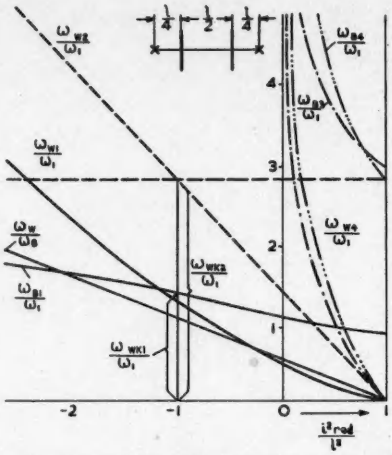



Fig. 5. **Frequency diagram of the system shown in Fig. 4.**

For the non-rotating shaft with the masses $m_1$ and $m_2$, and the moments of inertia $\Theta_{y1}$ and $\Theta_{y2}$, the natural angular velocity $\omega_B$ is determined, as already stated, by means of eqs. (1), (2) and (3). For the first-order vibration, all the deflections and obliquities should be added together according to eqs. (1). In the third-order vibration, the values of $\xi_1$ and $\xi_2$ are situated on the same side; the deflections $\xi_{13}$ and $\xi_{14}$ contributed by the moments $M_1$ and $M_2$ are, however, in opposite directions. If the shaft were rotated at an infinitely great speed ($\omega_W = \infty$), then infinitely large gyroscopic forces would occur, or, in the case of a finite vibration, the two shafts would not change their direction ($\varphi = 0$). In the case of the second- and fourth-order vibrations, the deflections $\xi_1$ and $\xi_2$ of the two masses are not directed towards the same side. The shaft then has a node situated between $m_1$ and $m_2$.

Therefore, the point $\Theta_{red_1} = \Theta_{y1}$ will correspond on the vertical axis to the point $\omega_W/\omega_B = 0$, and the points $\omega_{B2}$, $\omega_{B3}$, and $\omega_{B4}$.* For each value of $\Theta_{red_1}$ there are thus four different natural frequencies.

According to eq. (3a) these also correspond to various angular velocities $\omega_W$ if $\Theta_{red}$ is regarded as constant. This means that the running speed $\omega_W$ corresponds to various values $\omega_{B1}$, $\omega_{B2}$ and, therefore, to various values $\Theta_{red_1}$ and $\Theta_{red_2}$. For a given value of $\Theta_{red_1}$ we have the ratios $\omega_{W1}/\omega_{B1} = \omega_{W2}/\omega_{B2} = \omega_{W3}/\omega_{B3} = \omega_{W4}/\omega_{B4}$.

By means of eqs. (1), (2), and (3), we now calculate the points on the ordinate axis which correspond to a given value of $\Theta_{red_1}$ (for instance, assuming $\Theta_{red_1} = \Theta_{y1}/2$). From eq. (3a) the value $\omega_W/\omega_B$ corresponding to $\Theta_{red_1}$ is calculated, and then introducing this value into eq. (5b) we obtain the corresponding value for $\Theta_{red_2}$. Thus, all the points can be determined for a shaft with the masses $m_1$, $m_2$, $\Theta_{red_1}$ and $\Theta_{red_2}$, and the same calculation can then be subsequently applied for another selected value of $\Theta_{red_1}$ giving another $\Theta_{red_2}$.

When $\Theta_{red_1} = 0$ the ratio $\omega_W/\omega_B = \Theta_{y1}/\Theta_{x1}$ according to eq. (5a). For a thin disk for which $\Theta_{y1} = \Theta_{x1}/2$ we have, in this special case $\omega_W = \omega_B/2$. If the second moment of inertia is also constituted by a thin disk, then by virtue of eq. (5b) we also have $\Theta_{red} = 0$.

When the shaft rotates at still higher speeds, the inertias $\Theta_{red_1}$ and $\Theta_{red_2}$ assume negative values. A moment of inertia is "negative" when it stores nega-

*The order of the vibration is denoted by Roman numeral subscripts when only one subscript is used (e.g., $\omega_{III}$), and by Arabic numerals when used in conjunction with other subscripts (e.g., $\omega_{B3}$, $\omega_{W3}$, or $\omega_{WK1}$).

tive kinetic energy while swinging through the neutral or equilibrium position, and thereby draws this energy from the deformation energy of the shaft while swinging further along towards its extreme position. A negative moment of inertia is, therefore, equivalent to an auxiliary spring, which is unstrained in the mid-position of the vibration and subjected to maximum stresses in the extreme position, with the result that it then removes most of the energy of deformation of the shaft. Negative moments of inertia connected to a positive spring cannot perform real vibrations. Therefore, as soon as $\Theta_{red_1}$ and $\Theta_{red_2}$ become negative, the only two types of vibration which are possible are those which would occur in the case of masses concentrated at two points. During the vibration, additional bending moments are transmitted from the moments of inertia to the shaft, but these bending moments only affect the two natural angular frequencies $\omega_I$ and $\omega_{II}$ without generating new orders of vibration. Positive moments of inertia decrease the natural frequency $n_1$ as compared with its value for the arrangement with point-masses along the shaft, and negative moments of inertia, i.e., auxiliary springs, increase the value of $n_I$.

Critical speeds occur when $n_W = n_B$. From eqs. (5a) and (5b) it follows, however, that in this case $\Theta_{red_1}$ and $\Theta_{red_2}$ can only have positive values if $\Theta_y > \Theta_x$, that is, if the shaft is provided with long cylindrical masses instead of narrow disks.

If the ratios $\Theta_{yn}/\Theta_{xn}$ have different values for each of the $n$ masses considered, then the reduced moments of inertia $\Theta_{red_1}$, $\Theta_{red_2}$, etc., will be equal to zero at different speeds. However, if the shaft has narrow disks for which $\Theta_y = \Theta_x/2$, then, according to eq. (3), all the values $\Theta_{red}$ will be equal to zero for a shaft speed $n_W$ equal to $n_B/2$. But $\Theta_{red} = 0$ signifies that at this point $\Theta_{red}$ corresponds to an infinitely large value $n_{B_2}$. Therefore, the natural frequency $n_B$ passes at the point $\Theta_{red} = 0$ from a real value to an imaginary value, via infinity, when drawing the curve $n_B = f(\Theta_{red})$.

For speeds $\omega_W$ which are lower than $\omega_B/2$, there are twice as many natural frequencies as there are masses on the shaft, whereas for frequencies $\omega_W$ higher than $\omega_B/2$ there are $n$ natural frequencies for $n$ masses. In the case of long cylindrical masses for which $\Theta_y > \Theta_x$, the reduced moment of inertia $\Theta_{red}$ will only vanish when $\Theta_W/\Theta_B > 1$. This means that for shafts with such long cylindrical masses there will be twice as many critical speeds $n_{WK}$ as there are masses on the shaft.

## 4. SYMMETRICAL SYSTEM WITH TWO DISKS, EACH OF WHICH IS SITUATED AT A DISTANCE $l/4$ FROM THE END BEARINGS

The elastic lines of the four modes of vibration can be determined in a particularly simple manner for a symmetrical arrangement with two equal masses $m_1 = m_2$ and $\Theta_1 = \Theta_2$, the two masses being situated at a distance $l/4$ from their corresponding supports. It will be noted first of all that the elastic line of the 1st order vibration with the frequency $\omega_{I_1}$ (motion of type 1) has the shape shown in Fig. 6 assuming concentrated point-masses ($i_1 = i_2 = 0$).

A similar symmetrical shape of the elastic line is given in Fig. 7 for massless moments of inertia ($m_1 = m_2 = 0$), if we assume that the moments of inertia deflect in opposite directions (bending moments $M_1 = -M_2$). The corresponding frequency of the 1st order will be denoted by $\omega_{I_2}$ (motion of type 2).

If we add together the elastic lines of the vibrations given in Figs. 6 and 7 in such a way that both frequencies will have the same value, we then obtain the vibration frequency $\omega_I$ of the combined vibration. As the resultant vibration is obtained from two types of vibration by adding the deflections $\xi$ or $\varphi$, we have $\omega_I < \omega_{II}$ and $\omega_I < \omega_{I_2}$.

The 3rd order frequency $\omega_{III}$ of the entire arrangement is obtained by subtracting the elastic lines of Figs. 6 and 7, hence $\omega_{III} > \omega_{II}$ and $\omega_{III} > \omega_{I_2}$.

Elastic lines can be determined in a similar manner in the case where the two masses vibrate in two opposite directions ($P_1 = -P_2$), while the bending moments are in the same direction ($M_1 = M_2$). The elastic line with the frequency $\omega_{III}$ for point-masses is given in Fig. 8, and that for the frequency $\omega_{II_2}$ for massless moments of inertia is shown in Fig. 9. As a result of the assumption made that the masses are symmetrically arranged at distances $l/4$ from their respective bearings. the masses $m_1$ and $m_2$ will not change their direction or be twisted under the action of the forces $P_1 = -P_2$ in the vibration of Fig. 8. The frequency $\omega_{III}$ is, therefore (independent of the value of $\Theta_{red}$) also the vibration frequency $\omega_{II}$ of the entire system. In the frequency diagram of Fig. 5 we have, therefore, represented $\omega_{B_3} = f(\Theta_{red})$ by a straight line parallel to the $X$-axis, while $\omega_{W_3} = f(\Theta_{red})$ is a straight line which has the value $\omega_{W_3} = 0$ at the point $\Theta_{red_1} = \Theta_{y_1}$ and which intersects the curve $\omega_{B_3} = f(\Theta_{red_1})$ at a point situated above the intersection of the 1st order curves.

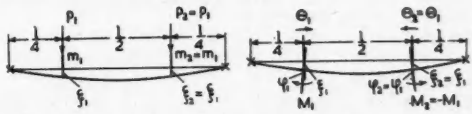
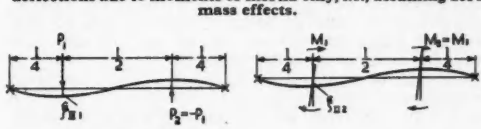



Fig. 6. (left) **Elastic line of a symmetrical system with deflections due to masses only. Masses are assumed to have zero moments of inertia.**

Fig. 7. (right) **Elastic line of a symmetrical system with deflections due to moments of inertia only, i.e., assuming zero mass effects.**




Figs. 8 and 9. **Elastic lines of the systems shown in Figs. 6 and 7, with a node between the masses.**

The fourth-order frequency $\omega_{IV}$ is finally obtained by adding together the elastic lines of Figs. 8 and 9. In this addition $\omega_{II_2}$ will be changed, since no change of angle is connected with Fig. 8. When $\omega_{III} > \omega_{II_2}$, the addition requires that the frequency $\omega_{III}$ be decreased to such an extent that the vibration frequency of the masses will be equal to the vibration frequency $\omega_{II_2}$ of the moments of inertia:

$$\omega_{II_2} = \omega_{IV} = \omega_{III}\sqrt{(\xi_{III} - \xi_{II_2})/\xi_{III}} \quad .. \quad (6a)$$

This equation gives the value of $K = M_1/(P_1\, l_1)$ which is used to plot Figs. 8 and 9 before adding the two curves.

When $\omega_{III} < \omega_{II_2}$, the two elastic lines must be added in accordance with the relation

$$\omega_{II_2} = \omega_{IV} = \omega_{III}\sqrt{(\xi_{III} + \xi_{II_2})/\xi_{III}} \quad .. \quad .. \quad (6b)$$

When $\omega_{III} = \omega_{II_2}$ the values of the vibration frequencies of the second and fourth orders coincide:

$$\omega_{III} = \omega_{II_2} = \omega_{II} = \omega_{IV} \quad .. \quad .. \quad .. \quad (6c)$$

In the limiting case $M = 0$, there is no obliquity in the position of the moments of inertia, so that the second and fourth order frequencies are equal.

In Fig. 5 $\omega_W/\omega_B = f(\Theta_{red_1})$ is a straight line. The critical speeds $\omega_{W_1} = \omega_{B_1}$ and $\omega_{W_2} = \omega_{B_2}$ are situated on the perpendicular through the point $\omega_W/\omega_B = 1$. The value $\Theta_{red_1}$ corresponding to $\omega_W/\omega_B = 1$ is equal to $-\Theta_{y_1}$ according to eq. (5a) when $\Theta_x = 2\,\Theta_y$. The frequency $\omega_{B_2}$ is independent of the shaft speed $\omega_W$ and, therefore, also independent of $\Theta_{red_1}$. The curve $\omega_{B_2} = f(\Theta_{red_1})$, as shown in Fig. 5, is a straight line parallel to the $X$-axis. Neither $\omega_{W_3}$, $\omega_{B_3}$, nor $\omega_{W_4}$, $\omega_{B_4}$ intersect one another in the real domain, since

$\theta_y < \theta_x$. In Fig. 5 it has been arbitrarily assumed that $\omega_{B_3} = \omega_{B_4}$ for the non-rotating shaft, and that $\omega_{B_2} < \omega_{B_3}$.

## 5. FIRST AND SECOND-ORDER VIBRATIONS OF THE SYMMETRICAL ARRANGEMENT WITH TWO MASSES

As a generalization of the previous case we shall now consider the first-order vibration of a symmetrical arrangement in which the distance of the masses from the end bearings may have any value between $l/2$ and 0. The distance $s$ of the two masses relatively to one another, therefore, varies between 0 and $l$. The frequency $\omega_1$ of the system for which $s = 0$ can be readily determined, by assuming $m_1$ and $m_2$ to be equal concentrated masses:

$$\omega_1^2 = (48EI)/(2l^3 m_1) \quad .. \quad .. \quad (7)$$

$\omega_1$ is thus equal to the frequency of a shaft which has a single mass equal to $2m_1$ situated at its centre.

When $s = l$, we have $\omega_1^2 = \infty$, since the deflections corresponding to the forces $P_1$ and $P_2$ acting on the masses $m_1 = m_2$ are zero.

The vibration frequency $\omega_2$ of the first order for the arrangement with massless moments of inertia ($m_1 = m_2 = 0$) is

$$\omega_2^2 = (2EI)/(mi^2 s) \quad .. \quad .. \quad .. \quad (8)$$

When $s = 0$, $\omega_2^2 = \infty$; and when $s = l$, $\omega_2^2 = (2EI)/(ml\,i^2)$. Moreover, when $s = 0$, $\omega_1^2 = (24EI)/ml^3$); and when $s = l$, $\omega_1^2 = \infty$.* When $s$ is varied between these limits, there is, of course, also a particular value for which $\omega_1 = \omega_2$. This shows, therefore, that, in general, it is essential to take both $\omega_1$ and $\omega_2$ into account when determining the natural vibration frequency of a shaft with several masses.

Furthermore, this example shows that in a two-mass arrangement performing a motion of the 1st type, the first-order vibration is with the two masses vibrating in the same direction, while the second-order vibration is for masses with opposite phases. In principle, the first-order vibration is at a lower frequency than the second-order vibration.

In the case of motion of the 2nd type, it is also possible to differentiate between vibrations of the first and second orders. The first-order vibration is, as previously, the vibration in which the deflections $\varphi_{11}$ and $\varphi_{12}$ at position 1, and $\varphi_{21}$ and $\varphi_{22}$ at position 2, due to the bending moments $M_1$ and $M_2$, are additive. The deflections are added when the bending moments are in opposite directions. In the second-order vibration $\omega_{II_2}$ the angular deflections are subtracted from one another ($\varphi_{11} - \varphi_{12}$, and $\varphi_{22} - \varphi_{21}$), and the bending moments are in the same direction. The frequency $\omega_{II_2}$ is only higher than $\omega_{I_2}$ if $s$ is not too small relatively to $l$. There will, of course, also be a value of $s$ for which $\omega_{II_2} = \omega_{I_2}$. When $s \to 0$, $\omega_{I_2} = \infty$, and there only remains the frequency $\omega_{II_2}$. For a shaft of normal dimensions with normal size masses, $\omega_{II_2}$ is greater than $\omega_{I_1}$. Thus, in the case of a shaft which only has a single mass, it is justifiable by way of approximation to assume that the mass is concentrated at a single point, since the vibration of the first order of a motion of the 1st type is combined with the second-order vibration of the motion of the 2nd type. When, however, several masses are on the shaft, there is no valid reason for neglecting the motion of the 2nd type in the approximation, since then the first-order vibration of the motion of the 1st type combines with the first-order vibration of the motion of the 2nd type, in order to give the first-order vibration of the entire system. These two types of motion, however, have low frequencies of the same order of magnitude, so that neither can be regarded as predominant.

*According to eq. (40) of Technical Report No. 4 of the Wöhler Institute.

The first-order vibration is defined as the vibration for which the deflections due to forces $P_1$ and $P_2$, and the moments $M_1$ and $M_2$, are additive. The second-order vibration is obtained by subtracting the deflections.

In the first-order vibration for motion of the 2nd type, the bending moments are directed in opposite directions, and there is no node in the system, whereas the second-order vibration has a node between the masses and moments in the same direction. In this respect, the conditions are, therefore, similar to those for the first- and second-order vibrations of concentrated masses. A fundamental difference, however, is that the frequency of the second-order vibration can in the case of motion of the second type, not only be higher, but also equal, or even lower than that of the first-order vibration.

The frequency of the motion of the 1st type in the arrangement with a single mass between the two bearings is, thus, generally much lower than the frequency of the motion of the 2nd type for this arrangement. This is because in the transition from the two-mass arrangement to the single-mass arrangement, for which $s = 0$, we obtain $\omega_1$ from the first-order vibration of the two-mass arrangement, whereas the motion of the 2nd type of the arrangement with a single moment of inertia between the bearings is obtained from the second-degree vibration of the arrangement with two moments of inertia on the shaft, for a distance $s$ equal to zero between the moments of inertia.

The overhung shaft with a single mass can also be regarded as having been obtained from an arrangement consisting of a cantilever shaft carrying two masses situated at distances $l$ and $l + s$ from the clamped end. The motion of the 2nd type of this arrangement can again be a first-order vibration without a node between the masses, or a second-order vibration with a node between the masses. The system becomes a single-mass system when we put $s = 0$. The frequency of the motion of the 2nd type for this arrangement is equal to $\omega_{I_2}$, and it is evident that the frequency of vibration of an overhung shaft with one mass, for which neither $m$ nor $i$ is equal to zero, is obtained by adding two motions which have the same frequency.†

In principle, we can, therefore, regard the resultant vibration obtained by summation of the two motions of the first order, i.e. of motions $I_1$ and $I_2$, as the first-order vibration of the entire arrangement. The second-order vibration of the entire arrangement is then obtained by summation of the motions of the second order of types 1 and 2. The third-order vibration is the difference between the motions of the first order of types 1 and 2, and the fourth-order vibration is the difference between the second-order vibrations of types 1 and 2. The first and second order vibrations of the two-mass system can be called the "principal vibrations," since they also occur when the distance between the two masses becomes infinitely small. In this case, the principal vibrations become the first and second-order vibrations of the shaft with a single mass.

In brief, it can be said in considering the vibration of a shaft with a single mass between two bearings, that the motion of the 1st type is decisive for its lowest vibration frequency, whereas for the overhung shaft or the shaft carrying several masses, the motions of the 1st and of the 2nd type are of equal importance. The second-order vibration of an arrangement with a single mass between the two bearings (characterized by a nodal point between the bearings), can be at a frequency which may have a higher, equal, or even a lower value, than the frequency of the first order (without a node between the bearings), according to the dimensions of the system.

†The Wöhler Institute report already mentioned gives the conditions for which the motions of the 1st and 2nd type will have equal frequencies.

# The Dynamic Capacity of Rolling Bearings

By G. LUNDBERG and A. PALMGREN. (From *Acta Polytechnica*, Mechanical Engineering Series, Vol. 1, No. 3, Issue 7, 1947, 50 pages, 22 illustrations.)

(*Continued from November, 1948, issue.*)

Let $u$ = the number of stress cycles per revolution, and $L$ = life in millions of revolutions, so that $N = uL$. Introducing the raceway diameter $D_a$ and the rolling body diameter $D_a$, and the abbreviations

$$\left.\begin{array}{l} c+h-1=2\beta_1,\ c-h-1=2\beta_2, \\ c-h+1=2\beta_3,\ 2c+h-2=3\beta_4, \\ c-h+2=3\beta_5, \end{array}\right\} \quad (32)$$

we then obtain from eqs. (29) to (31)

$$\log\frac{1}{S} \propto T^c\,\zeta^{1-h}\left[\frac{E_0\,D_a\,\Sigma\rho}{3\,\mu\,\nu^2}\right]^{\beta_1}\times\left[\frac{D_a}{a}\right]^{\beta_2}\times \left[\frac{Q}{D_a^{\,2}}\right]^{\beta_3} D_n\,D_a^{\,2-h}\,u^e\,L^e \quad .. \quad (33)$$

this formula being valid for both point and line contact. For point contact, eq. (31) can be introduced into eq. (33), so that

$$\log\frac{1}{S} \propto T^c\,\zeta^{1-h}\,\mu^{1-c}\,\nu^{-2\beta_1}\left[\frac{E_0\,D_a\,\Sigma\rho}{3}\right]^{\beta_4}\left[\frac{Q}{D_a^{\,2}}\right]^{\beta_5} \times D_n\,D_a^{\,2-h}\,u^e\,L^e \quad .. \quad (34)$$

If the major axis of the pressure ellipse is equal to the length of the roller ($2a = l_a$), point contact is transformed into line contact. The load in this case still produces an elliptical pressure area and an ellipsoidal pressure distribution within the contact area. At higher loads a levelling of pressure occurs, and according to Hertz's theory, an even distribution of pressure in the lengthwise direction of the contact area occurs for a load corresponding to $2a = 1{\cdot}5\,l_a$, while for higher loads edge pressure occurs.

Since $b/a$ is small, we can introduce the value

$$\lim_{b/a\,\to\,0}\mu\nu^2 = \frac{2}{\pi} \quad .. \quad .. \quad (35)$$

If $T_0$ and $\tau_0$ are the values of $T$ and $\tau$ corresponding to $b/a = 0$, then for line contact

$$\log\frac{1}{S} \propto T_0^{\,c}\,\zeta_0^{1-h}\left[\frac{\pi\,E_0\,D_a\,\Sigma\rho}{6}\right]^{\beta_1}\left[\frac{4\,D_a}{3\,l_a}\right]^{\beta_2} \times\left[\frac{Q}{D_a^{\,2}}\right]^{\beta_3} D_n\,D_a^{\,2-h}\,u^e\,L^e \quad .. \quad (36)$$

Eqs. (34) and (35) can also be written

$$\frac{Q}{D_a^{\,2}}L^{e/\beta_5} = A_1\,\Phi\,D_a^{\,(3-h)/\beta_5} \quad .. \quad (37)$$

for point contact, and

$$\frac{Q}{D_a^{\,2}}L^{e/\beta_3} = B_1\,\Psi\,D_a^{\,(c-3h+5)/\beta_3}\,l^{\,\beta_2/\beta_3} \quad .. \quad (38)$$

for line contact.

If $T_0$, $\tau_0$, and $T_1$, $\tau_1$ are functions of $T$ and $\zeta$, for $b/a = 0$ and 1, respectively, then

$$\Phi = \left[\left(\frac{T}{T_1}\right)^c\left(\frac{\zeta_1}{\zeta}\right)^{h-1}\frac{(D_a\,\Sigma\rho)^{\beta_4}}{\mu^{c-1}\,\nu^{2\beta_1}}\,\frac{D_n}{D_a}\,u^e\right]^{-\frac{1}{\beta_5}} \quad (39)$$

$$\Psi = \left[(D_a\,\Sigma\rho)^{\beta_1}\,\frac{D_n}{D_a}\,u^e\right]^{-\frac{1}{\beta_1}}$$

and

$$B_1 = \left[\left(\frac{3}{4}\right)^{\beta_2}\times\left(\frac{\pi}{2}\right)^{\beta_1}\times\left(\frac{T_1}{T_0}\right)^c\times\left(\frac{\zeta_0}{\zeta_1}\right)^{h-1}\times \left(\frac{E_0}{3}\right)^{\beta_2/3}\times A_1^{\beta_5}\right]^{\frac{1}{\beta_3}} \quad .. \quad (40)$$

If $Q_c$ is the rolling body load for the basic dynamic capacity of the bearing, i.e. for $L = 1$, then according to eqs. (36) and (37)

$$Q_c = A_1\,\Phi\,D_a^{\,(\beta_4-1)/\beta_5} \quad .. \quad .. \quad .. \quad (41)$$

for point contact, and

$$Q_c = B_1\,\Psi\,D_a^{\,(\beta_1-1)/\beta_3}\,l_a^{\,\beta_2/\beta_3} \quad .. \quad .. \quad (42)$$

for line contact.

Since the rolling body load is proportional to the bearing load, $Q_c$ and $Q$ are proportional to the basic dynamic capacity $C$ and the actual bearing load $F$, respectively. Hence from eqs. (37), (38), (41) and (42) we have

$$L = (C/F)^p \quad .. \quad .. \quad (43)$$

where $p = \beta_5/e$ for point contact, and $p = \beta_3/e$ for line contact.

For radial bearings, the load on a rolling element depends on the rolling element's position. In this case $Q_c$ and $Q$, respectively, represent mean values of the load on the rolling elements, viz. the equivalent rolling element load.

## 4. DETERMINATION OF EXPONENTS IN THE LIFE FORMULA

The exponents $c$, $h$, and $e$, in eqs. (32), (34) and (36), can be regarded as material characteristics and be determined experimentally by finding : (i) the value of $p$ in eq. (43); (ii) the exponent of $D_a$ in eq. (41) (dependence of basic dynamic capacity on bearing size) for point contact, and (iii) the dispersion of bearing life under uniform conditions of operation.

Fig. 4 shows a series of tests with a total of 270 self-aligning ball bearings (SKF 1309) and 180 deep groove ball bearings (SKF 6309), with straight lines for log $1/S$ having the slope $e = 10/9$. Fig. 5 shows a test series for spherical roller bearings (SKF I-37906) where $e = 1{\cdot}5$. Different SKF roller bearing series of different dimensions, made up of 227 bearings, also gave a value for $e$ varying between 1·1 and 2·1. The

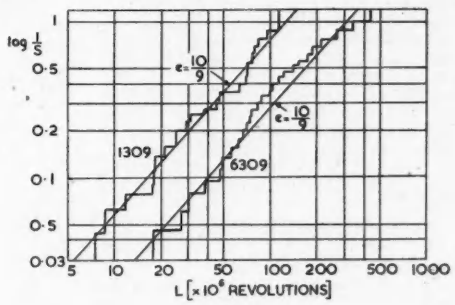



Fig. 4

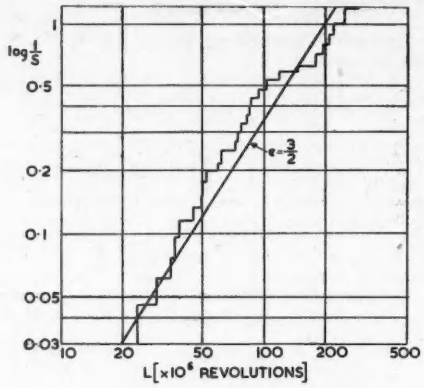



Fig. 5

probable values of $e$ as a result of these tests are: $e = 1$ for ball bearings, and $e = 1{\cdot}5$ for roller bearings.

The exponent $p$ for ball bearings was determined from a series of tests with bearings SKF 1309 and 6309. In Fig. 6, each point represents the mean life of a group of 30 bearings. The solid line corresponds to the exponent $p = 3$. For roller bearings similar tests are not yet available. The indications are, however, that the value of $p$ may be somewhat higher for roller bearings than for ball bearings.

For the determination of the dependence of the basic dynamic capacity on the size of bearings, the results from tests on about 1500 bearings are given in Fig. 7. The figure shows that within the test range, from $D_a = 1{\cdot}5$ mm to $D_a = 25$ mm, the basic dynamic capacity $C = F\,L^{1/3}$ is approximately proportional to $D_a^{1{\cdot}8}$, corresponding to the solid line. The results obtained are summarized in Table II.

TABLE II.

| | | | |
|---|---|---|---|
| | $e$ .. .. | 10/9 | 3/2 |
| | $c$ .. .. | $10\frac{1}{3}$ | $10\frac{1}{3}$ |
| | $h$ .. .. | $2\frac{1}{3}$ | $2\frac{1}{3}$ |
| *Point contact*: | | | |
| | $p$ .. .. | 3 | — |
| Exponent of | $D_a$ .. .. | 1·8 | 1·8 |
| *Line contact*: | | | |
| | $p$ .. .. | 4·05 | 3 |
| Exponent of | $D_a$ .. .. | 1·07 | 1·07 |
| Exponent of | $l_a$ .. .. | 0·78 | 0·78 |

There would be great practical difficulties if the exponent $p$ were taken as different for point and line contact. Various roller bearings designs are so constructed that pronounced point contact holds at low loads, and pronounced line contact at high loads. The value of $p$ would then vary according to the type of contact decisive for dynamic capacity. It seems, thus, justifiable to use for all bearings a common value $p = 3$. If this possibly means undervaluing roller bearings with nominal line contact at low loads, the additional safety in the calculation of such bearings appears reasonable, in view, for instance, of the uncertainty regarding the pressure distribution along the contact surface.

In the following sections, the value $p = 3$ will be used for all calculations, for both point and line contact.

Then eqs. (39) to (42) give

$$Q_c = A_1\,\Phi\,D_a^{1{\cdot}8}, \qquad .. \quad (44)$$

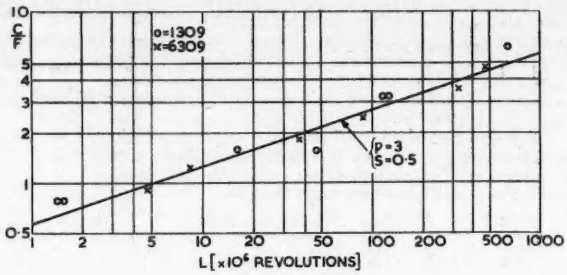



Fig. 6

for point contact, and

$$Q_c = B_1\,\Psi\,D_a^{1{\cdot}07}\,l_a^{0{\cdot}78}, \qquad .. \quad (45)$$

for line contact, where

$$\left.\begin{aligned} \Phi &= \left(\frac{T_1}{T}\right)^{3{\cdot}1}\left(\frac{\zeta}{\zeta_1}\right)^{0{\cdot}4}\frac{\mu^{2{\cdot}8}\,\nu^{3{\cdot}5}}{(D_a\,\Sigma\rho)^{2{\cdot}1}}\left(\frac{D_a}{D_n}\right)^{0{\cdot}3}u^{-1/3} \\ \Psi &= (D_a\,\Sigma\,\rho)^{-35/27}\left(\frac{D_a}{D_n}\right)^{2/9}u^{-1/3}, \\ B_1 &= 3{\cdot}517\,A_1^{20/27}. \end{aligned}\right\} \quad (46)$$

The accepted values of the exponent $e$ imply that the median life for ball bearings is 5·4 times and for roller bearings 3·5 times the life which is exceeded by 90 per cent of all bearings in a large test series.

## 5. DISTRIBUTION OF LOAD AMONG THE ROLLING ELEMENTS

### (1) *Single-Row Bearings*

If, while remaining parallel, the bearing rings are displaced relatively to each other over a distance $\delta_a$ in the axial direction, and $\delta_r$ in the radial direction (Fig. 8), then the approach of the rings at an angular distance $\psi$ from the most heavily loaded element is

$$\delta(\psi) = \delta_a \sin\alpha + \delta_r \cos\psi \cos\alpha \qquad .. \quad (47)$$

In bearings without internal looseness, the elastic compression in the contacts of the most heavily loaded rolling body, for which $\psi = 0$, is

$$\delta_{max} = \delta_a \sin\alpha + \delta_r \cos\alpha \qquad .. \quad (48)$$

Then from eqs. (47) and (48)

$$\left.\begin{aligned} \delta(\psi) &= \delta_{max}\left[1 - \frac{1}{2\epsilon}(1 - \cos\psi)\right] \\ \epsilon &= \frac{1}{2}\left(1 + \frac{\delta_a \tan\alpha}{\delta_r}\right) \end{aligned}\right\} \quad .. \quad (49)$$

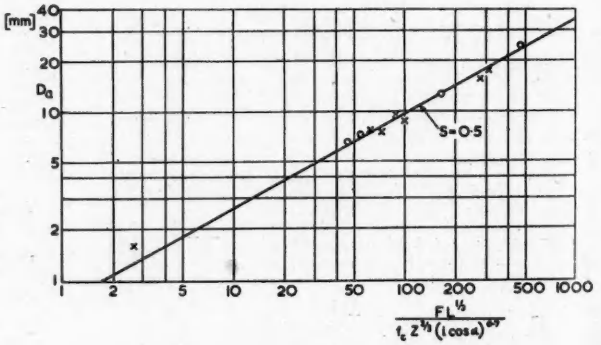



Fig. 7

where $\epsilon$ = a parameter which depends on the loaded zone (Fig. 8). The loaded zone covers a part of the periphery or the entire circumference, according as $\epsilon < 1$ or $\epsilon \geq 1$. If $\epsilon = 1/2$, then $\delta_a = 0$, and the displacement of the bearing rings is entirely radial, while if $\epsilon = \infty$ the displacement is completely axial and all rolling elements are equally loaded.

According to Hertz's theory, the rolling element loads for point contact are proportional to $\delta\,(\psi)^{3/2}$. For line contact, the rolling element loads are approximately proportional to $\delta\,(\psi)$. Then

$$Q\,(\psi) \cong Q_{max}\left[1 - \frac{1}{2\,\epsilon}\,(1 - \cos\,\psi)\right]^t \quad .. \quad (50)$$

where $t = 3/2$ for point contact, and $t = 1$ for line contact.

For single-row bearings, the radial and thrust components of the load are found from the equations

$$\left.\begin{aligned} F_r &= \Sigma Q\,(\psi)\cos\alpha\cos\psi \cong Z Q_{max}\cos\alpha\,\mathcal{J}_r \\ F_a &= \Sigma Q\,(\psi)\sin\alpha \cong Z Q_{max}\sin\alpha\,\mathcal{J}_a \end{aligned}\right\} \quad (51)$$

where $\mathcal{J}_r$ and $\mathcal{J}_a$ are the radial and thrust integrals introduced by Sjövall :

$$\left.\begin{aligned} \mathcal{J}_r\,(\epsilon) &= \frac{1}{2\,\pi}\int_{-\psi_0}^{+\psi_0}\left[1 - \frac{1}{2\epsilon}\,(1 - \cos\psi)\right]^t \cos\psi\,d\psi \\ \mathcal{J}_a\,(\epsilon) &= \frac{1}{2\,\pi}\int_{-\psi_0}^{+\psi_0}\left[1 - \frac{1}{2\epsilon}\,(1 - \cos\,\psi)\right]^t d\psi \end{aligned}\right\} \quad (52)$$

where $t = 3/2$ and $t = 1$ for point and line contact, respectively, and $\psi_0$ is equal to one half of the loaded arc.

(2) *Double-Row Bearings*

Denoting the rows of a two-row bearing by the indices I and II, then for parallel displacement of the bearing rings, for bearings without internal clearance

$$\delta_{aI} = -\delta_{aII} = \delta_a;\ \delta_{rI} = \delta_{rII} = \delta_r \quad .. \quad (53)$$

and according to eqs. (48) and (49), as long as both rows are loaded:

$$\delta_{maxII}/\delta_{maxI} = \epsilon_{II}/\epsilon_I;\ \epsilon_I + \epsilon_{II} = 1,$$

and if only row *I* is loaded:

$$\epsilon_I \geq 1;\ \epsilon_{II} = 0.$$

The greatest rolling element loads in each row are obtained from the relations

$$Q_{maxII}/Q_{maxI} = (\epsilon_{II}/\epsilon_I)^{3/2},$$

and

$$Q_{maxII}/Q_{maxI} = \epsilon_{II}/\epsilon_I,$$

for point and line contact respectively, $Q$ designating the rolling body load.

For double-row bearings, the radial and thrust components of load are found from

$$\left.\begin{aligned} F_r &= F_{rI} + F_{rII} = Z\cos\alpha\,Q_{maxI}\,\mathcal{J}_r \\ F_a &= F_{aI} - F_{aII} = Z\sin\alpha\,Q_{maxI}\,\mathcal{J}_a \end{aligned}\right\} \quad (54)$$

where

$$\left.\begin{aligned} \mathcal{J}_r &= \mathcal{J}_r\,(\epsilon_I) + [Q_{maxII}\,\mathcal{J}_r\,(\epsilon_{II})/Q_{maxI}] \\ \mathcal{J}_a &= \mathcal{J}_a\,(\epsilon_I) - [Q_{maxII}\,\mathcal{J}_a\,(\epsilon_{II})/Q_{maxI}] \end{aligned}\right\} \quad (55)$$

and $Z$ = number of rolling bodies per row.

In addition, for both single-row and double-row bearings, according to eqs. (52) and (54)

$$F_r \tan\alpha/F_a = \mathcal{J}_r/\mathcal{J}_a \qquad (*) \quad .. \quad (56)$$

(3) *Changes in Contact Angle with Load.*

For most bearing types the contact angle can be regarded as independent of load. For deep-groove and angular-contact ball bearings, however, the contact angle varies somewhat with the load, particularly if the nominal contact angle is small and if pure thrust loads are considered.

In Fig. 9, $A$ and $B$ are the profile centres of outer and inner rings. With an axial displacement $\delta_a$ of the inner ring profile centre from $B$ to $B_1$, the contact angle changes from $\alpha_0$ $\alpha$. Then $AB = r_i + r_e - D_a$ and $AB_1 \cong r_i + r_e - D_a + \delta_a \sin[(\alpha + \alpha_0)/2]$. Furthermore, $AB\cos\alpha_0 = AB_1\cos\alpha$.

Therefore

$$\frac{\cos\alpha_0}{\cos\alpha} = 1 + \frac{\delta_a}{r_i + r_e - D_a}\sin\frac{\alpha + \alpha_0}{2} \quad (57)$$

and $(\delta_{max}/D_a)^3 = c^3(Q_{max}/D_a^2)^2 = c^3F_a^2/(\mathcal{J}_a Z D_a^2 \sin\alpha)^2$

$$\delta_a \sin\frac{\alpha + \alpha_0}{2} = \delta_{max}\,[1 - (1/2\epsilon)], \quad .. \quad (58)$$

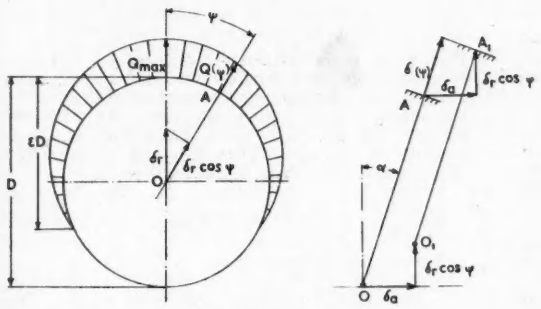



Fig. 8

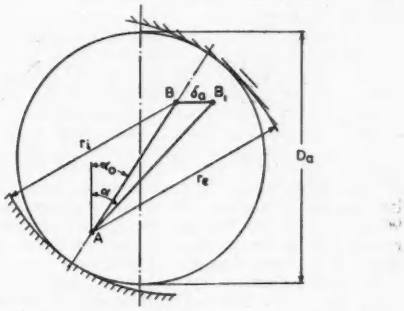



Fig. 9

where $c$ is a compression constant which depends on the conformity between rolling elements and bearing rings and on the modulus of elasticity. The curvature of the raceways in the direction of rolling has only a slight influence on the total compression $\delta_{max}$ at the two contacts, and can therefore be neglected. Then from Hertz's theory, with units in kg and mm: for $r_i/D_a = r_e/D_a = 0{\cdot}5175$, $0{\cdot}53$ and $0{\cdot}54$, we have $c = 0{\cdot}00201$, $0{\cdot}00227$, and $0{\cdot}00242$, respectively.

For bearings with different contacts at inner and outer rings, the mean value of $c$ can be introduced, corresponding to the values of the conformities for the inner and outer ring. Then, from eqs. (57) and (58):

(*) A table for the values for $\mathcal{J}_a$, $\mathcal{J}_r$, and $F_r$ tan $\alpha/F_a$ for point contact and line contact for single-row and double-row bearings is given in the original article.

$$\frac{\cos \alpha_0}{\cos \alpha} = 1 + \frac{c\, D_a}{r_i + r_e - D_a}\left(1 - \frac{1}{2\epsilon}\right)\left(\frac{F_a}{\mathcal{J}_a Z D_a^2 \sin \alpha}\right)^{2/3} \quad .. \quad (59)$$

for an arbitrary direction of load.

In reality the contact angle $\alpha_0$ is also affected, to some extent, by radial displacement and depends on the position of the rolling element on the periphery.

For deep-groove ball bearings with a nominal contact angle $\alpha_0 = 0$, eq. (59) gives the approximation

$$\sin \alpha \simeq \tan \alpha \simeq \left(\frac{2_c D_a}{r_i + r_e - D_a}\right)^{3/8}\left(1 - \frac{1}{2\epsilon}\right)^{3/8}\left(\frac{F_a}{\mathcal{J}_a Z D_a^2}\right)^{1/4} \quad .. \quad (60)$$

For thrust loading $\epsilon = \infty$ and $\mathcal{J}_a = 1$, so that with eqs. (59) and (60)

$$\frac{\cos \alpha_0}{\cos \alpha} = 1 + \frac{cD_a}{r_i + r_e - D_a}\left(\frac{F_a}{ZD_a^2 \sin \alpha}\right)^{2/3} \text{ if } \alpha_0 \neq 0 \quad (61)$$

$$\tan \alpha \simeq \left(\frac{2c\, D_a}{r_i + r_e - D_a}\right)^{3/8}\left(\frac{F_a}{ZD_a^2}\right)^{1/4} \text{ if } \alpha_0 = 0 \quad (62)$$

For deep-groove ball bearings with $r/D_a = 0{\cdot}5175$ it is found that

$$(2\, c\, D_a/[r_i + r_e - D_a])^{3/8} \simeq 4/9.$$

For angular-contact ball bearings (for which $r_i/D_a = 0{\cdot}5175$ and $r_e/D_a = 0{\cdot}53$), subjected to thrust load, the values in Table III are valid.

TABLE III.

| $\alpha_0$ | 30° | | 40° | |
|---|---|---|---|---|
| $Q/D_a^2$ | $F_a/(ZD_a^2)$ | cotan $\alpha$ | $F_a/(ZD_a^2)$ | cotan $\alpha$ |
| 0 | 0 | 1·733 | 0 | 1·192 |
| 0·3 | 0·1586 | 1·598 | 0·1981 | 1·137 |
| 1 | 0·5598 | 1·480 | 0·6803 | 1·077 |
| 2 | 1·178 | 1·372 | 1·399 | 1·022 |
| 3 | 1·833 | 1·296 | 2·142 | 0·981 |
| 4 | 2·515 | 1·237 | 2·904 | 0·948 |
| 5 | 3·220 | 1·188 | 3·681 | 0·919 |
| 6 | 3·943 | 1·147 | 4·472 | 0·895 |

## 6. THE BASIC DYNAMIC CAPACITY

The basic dynamic capacity for radial or thrust bearings is defined as that radial or thrust load which 90 per cent of the bearings can endure for one million revolutions under certain specified conditions of operation. For thrust bearings, these conditions are pure axial displacement of the bearing (central thrust bearing); for radial bearings, pure radial displacement of the bearing rings, and inner ring rotation relatively to the direction of load while the outer ring is stationary (fixed outer ring load).

The basic dynamic capacity $C$ of the bearings depends on the probability of fatigue in the different parts of the bearing, i.e. on the capacity $C_i$ of the inner ring contact, and the capacity $C_e$ of the outer ring contact.

(1) *Thrust Bearings*

For favourable operating conditions, a central thrust bearing load is distributed equally among the rolling elements. The stress amplitude caused by the load under rotation is thus independent of time and of the position of the contact in the raceway. Therefore

$$C_i = Q_{ci} Z \sin \alpha \quad .. \quad .. \quad .. \quad (63)$$

$$C_e = Q_{ce} Z \sin \alpha \quad .. \quad .. \quad (64)$$

where $Q_{ci}$ and $Q_{ce}$ are found from eqs. (44) and (45) for point and line contact respectively.

It can be shown that

$$\frac{1}{C^w} = \frac{1}{C_i^w} + \frac{1}{C_e^w} \quad .. \quad .. \quad (65)$$

The basic dynamic capacity of the bearing is then

$$\left.\begin{aligned} C &= g_c\, C_i \\ \text{where} \quad g_c &= [1 + (C_i/C_e)^w]^{-1/w} \end{aligned}\right\} \quad (66)$$

and $w = pe = 10/3$ and $9/2$ for point and line contact respectively. The variation of $g_c$ with $C_i/C_e$ is shown in Fig. 10.

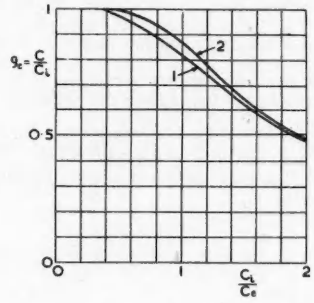



Fig. 10

(2) *Radial Bearings*

In a radial bearing, each point of the rotating *inner* ring passes through the loaded zone. All points on the inner ring are thus equally stressed, but the rolling element loads and the stress amplitudes vary periodically with the rotation. The approximate mean load is determined by the cubic mean value, in accordance with eqs. (22) and (23):

$$Q_{ci} = \sqrt[3]{\frac{1}{2\pi}\int Q^3(\psi)\, d\psi} = Q_{\max} \mathcal{J}_1 \quad .. \quad (67)$$

Each point on the stationary *outer* ring supports under rotation a constant rolling element load and, therefore, a constant stress amplitude; these depend, however, on the position of the point in the inner ring. The probability that the entire outer ring will endure is the product of the probabilities that the different outer ring portions will endure, so that from eq. (29)

$$\log \frac{1}{S_e} \propto \int Q^w(\psi)\, L^e \frac{D^e}{2} d\psi \propto Q_{ce}{}^w L^e \pi\, D_e \quad (68)$$

and the equivalent body load is

$$Q_{ce} = \left[\frac{1}{2\pi}\int Q^w(\psi)\, d\psi\right]^{1/w} = Q_{\max} \mathcal{J}_2 \quad .. \quad (69)$$

The resultant formulae (70) and (71) for the basic dynamic capacity are given below.

Inner ring $C_i = i^{k_i} K_i Q_{ci} Z \cos \alpha$ .. (70)
point contact $k_i = 0{\cdot}7$, $K_i = 0{\cdot}407$
line contact $k_i = 7/9$, $K_i = 0{\cdot}419$

Outer ring $C_e = i^{k_e} K_e Q_{ce} Z \cos \alpha$ .. (71)
point contact $k_e = 0{\cdot}7$, $K_e = 0{\cdot}389$
line contact $k_e = 7/9$, $K_e = 0{\cdot}367$.

$C_i = F_r$ is obtained from eqs. (51) and (64); $i$ = number of rows; $Z$ = number of rolling bodies per row; $Q_{ci}$ is determined from eqs. (44) and (45). $C_e = F_r$ is based on eqs. (51) and (69); $\mathcal{J}_1$ and $\mathcal{J}_2$ being obtained assuming a radial displacement of the bearing rings ($\epsilon = 0{\cdot}5$) so that $\mathcal{J}_1 = 0{\cdot}5625$ and $0{\cdot}5965$, and $\mathcal{J}_2 = 0{\cdot}5875$ and $0{\cdot}6814$, for point and line contact respectively, while $w$ is $10/3$ and $9/2$ respectively.

(3) *Effect of the Relative Conformity*

The principal curvatures of the bodies at the contact point are

$$\left.\begin{aligned} &\rho_{1\,\mathrm{I}} = 2/D_a,\ \rho_{2\,\mathrm{I}} = 1/R, \quad .. \\ &\rho_{1\,\mathrm{II}} = \pm\, 2\cos\alpha/D_n,\ \rho_{2\,\mathrm{II}} = -\,1/r \end{aligned}\right\} \quad (72)$$

where $r$ and $R$ are the raceway and roll body cross profile radii, and $D_n$ and $D_a$ the raceway and roll body diameters (Fig. 11). The + and — signs in $\rho_{1\,\mathrm{II}}$ are valid for raceways in the inner and outer rings, respectively.

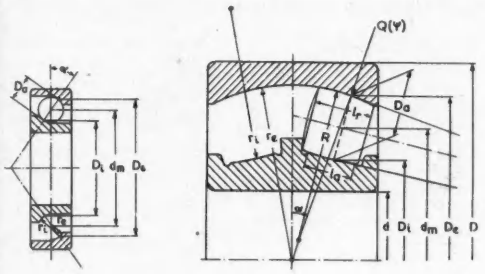



Fig. 11

From Hertz's theory for elasticity at contact, the shape $b/a = m$ of the contact area depends on

$$\mathbf{F}(m) = \frac{1}{\Sigma\rho}\left|\rho_{1\,\mathrm{I}} - \rho_{2\,\mathrm{I}} + \rho_{1\,\mathrm{II}} - \rho_{2\,\mathrm{II}}\right| \quad .. \quad (73)$$

$$= 1 - [2m^2(\mathbf{K} - \mathbf{E})]/[(1-m^2)\mathbf{E}] \quad .. \quad (74)$$

and thus $\nu = \mu m = \mu b/a = \sqrt[3]{2b\mathbf{E}/(\pi a)}$

where $\mathbf{K}$ and $\mathbf{E}$ are complete elliptic integrals of the first and second kind, respectively, with the modulus $1 - m^2$.

From eqs. (72) and (73), and with the abbreviations $M_1 = 1 \pm (D_a \cos\alpha/D_n)$ and $M_2 = 1 \mp (D\cos\alpha/D_n)$, $N = \frac{D_a}{2}\left(\frac{1}{R} - \frac{1}{r}\right)$, $T_1 = 1 \pm (D_a\cos\alpha/d_m)$ and $T_2 = 1 \mp (D_a\cos\alpha/d_m)$,

$$\left.\begin{aligned} \mathbf{F}(m)\,\frac{D_a}{2}\,\Sigma\rho &= M_1 - N \\ \frac{D_a}{2}\,\Sigma\rho &= M_1 + N \end{aligned}\right\} \quad (75)$$

The relation between the raceway diameter and the pitch diameter of the set of rolling bodies (Fig. 11) is obtained for the inner ring from

$$D_n = D_i = d_m - D_a\cos\alpha \quad .. \quad (76)$$

and for the outer ring from

$$D_n = D_e = d_m + D_a\cos\alpha \quad .. \quad (77)$$

$$\text{Then} \quad \left.\begin{aligned} [1 + \mathbf{F}(m)]\,\frac{D_a}{2}\,\Sigma\rho &= 2M_1 = 2/T_2 \\ [1 - \mathbf{F}(m)]\,\frac{D_a}{2}\,\Sigma\rho &= 2N \end{aligned}\right\} \quad (78)$$

$$\text{and} \quad \left.\begin{aligned} D_a\,\Sigma\rho &= 4/[(1 + \mathbf{F}(m))\,T_2] \\ \Omega &= [1 + \mathbf{F}(m)]/[1 - \mathbf{F}(m)] = NT_2. \end{aligned}\right\} \quad (79)$$

The shape of the pressure ellipse, that is, $b/a$, is obtained most simply from $\Omega$. The number of contact cycles per bearing revolution determined by

$$u = ZT_1/2 \quad .. \quad .. \quad (80)$$

where $Z$ = number of rolling bodies per row. Introducing the expressions given by eqs. (79) and (80) for $D_a\,\Sigma\rho$, and $u$, into eq. (46), the functions $\Phi$ and $\Psi$ are given by:

$$\Phi = 0{\cdot}06855\ \omega\,\frac{T_2^{1.8}}{T_1^{1/3}}\,(D_a/d_m)^{0.3}\,Z^{-1/3}$$

$$\Psi = 0{\cdot}5130\,\frac{T_2^{29/27}}{T_1^{1/3}}\,(D_a/d_m)^{2/9}\,Z^{-1/3} \quad .. \quad (81)$$

$$\omega = [1 + \mathbf{F}(m)]^{2.1}\left[\frac{T_1}{T}\right]^{3.1}\left[\frac{\zeta}{\zeta_1}\right]^{0.4}\mu^{2.8}\,\nu^{3.5}$$

Fig. 12 shows $\omega$, which depends on $m = a/b$ only, as a function of $\Omega$. Within the range corresponding to deep-groove ball bearings and roller bearings with point contact, $\omega$ can be replaced by

$$\omega \cong f = 1{\cdot}3\,\Omega^{-0.41} = 1{\cdot}3\,(NT_2)^{-0.41} \quad .. \quad (82)$$

For self-aligning ball bearings, the outer ring is usually decisive for fatigue and for that contact $m = 1$, $\Omega = 1$, and $\omega \simeq f = 1{\cdot}3$, using the approximate formula (82) in place of the theoretical value $\omega = 1$.

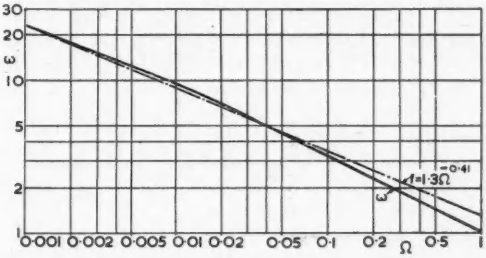



Fig. 12

The simplifying assumption so far made has been that the risks of flaking of the bodies in contact are both equally great, and this is approximately true for roller bearings. With ball bearings, the balls have the possibility of changing their axis of rotation, particularly when passing through an unloaded zone, and this explains why body flakings are less common in ball bearings, particularly in self-aligning ball bearings, where the width of the contact area is small. Nevertheless, it can be shown that the approximation given by eq. (80) is applicable to all types of bearings. Then from eqs. (44), (45), (81) and (82)

$$Q_c = A\,N^{-0.41}\,\frac{M_2^{1.39}}{M_1^{1/3}}\left(\frac{D_a}{d_m}\right)^{0.3} D_a^{1.8}\,Z^{-1/3} \quad (83)$$

for point contact, and

$$Q_c = B\,\frac{M_2^{29/27}}{M_1^{1/3}}\left(\frac{D_a}{d_m}\right)^{2/9} D_a^{29/27}\,l_a^{7/9}\,Z^{-1/3} \quad .. \quad (84)$$

for line contact, where $B = 10{\cdot}82\ A^{20/27}$. From the test series of Figs. 6 and 7 it is found that $A = 10$. For SKF bearing steel with a hardness of 61·7 — 64·5 Rockwell $C$ (750-825 Vickers), the average values are $A = 10$, $B = 60$ (units in kg and mm).

The capacities $C_i$ and $C_e$ of the inner and outer ring contacts can be calculated from eqs. (83) and (84), the upper signs in $M_1$ and $M_2$ being used for the inner ring and the lower signs for the outer ring. The overall capacity $C$ of the bearing is then obtained from eq. (64). For a bearing with point contact on one ring and line contact on the other, one can introduce the mean value $w = 4$ in eq. (64).

*(To be continued).*

ITALY

# The Compounding of Small Alternators

By G. SOMEDA. (From *L'Elettrotecnica*, Vol. 35, No. 3, March, 1948, pp. 118-119, 2 illustrations.)

THE construction of transportable electrical generating equipment for single- or three-phase alternating current supply at powers between 5 and 50 kVA was the subject of considerable development work during the war years. Generator sets of this type require an adequate voltage regulation for satisfactory operation, but, in general, voltage regulators are costly for the low-power machines under consideration, and also too delicate for non-specialized personnel.

Manufacturers therefore resorted to other means and generally converted the machine into a compound-wound design, with results similar to those usually obtained with d.c. machines working on this principle. Thus, an auxiliary converter unit was required to convert alternating current into direct current, and for this purpose, three systems have mainly been used : (1) static dry (copper-oxide or selenium) rectifiers ; (2) rotary converters ; (3) special exciters.

The regulation problem is in any case more complicated than that of the d.c. machine, since the voltage-drops to be compensated are not only functions of the load current but also of the power factor. The author developed a special exciter system for the Officine Pellizzari, and this has been tried out on a number of machines*.

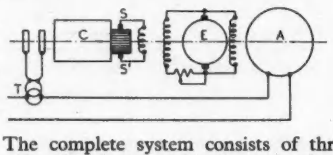



Fig. 1.

The complete system consists of three units (see Fig. 1): the alternator *A*, the co-axial exciter *E*, and the converter *C* supplied from the series transformer *T*. The converter is of the usual type with a simple armature wound for the same number of poles as the alternator *A* and provided with a ring closing the magnetic circuit. This ring consists of a pack of laminations mounted directly on the armature and rotating with it. The machine is thus without a stator, and this simplified construction is possible because of the very low powers involved which make the commutation problem unimportant. As all three machines are mounted on the same shaft, at a given speed of rotation the d.c. voltage collected at the brushes *S* and *S'* will depend on the current supplied by the alternator and on the power factor of the load.

For a given phase angle of the current generated, there is a position of the brushes *S*, *S'*, for which the voltage obtained will be at a maximum, and another position at right angles to it, for which the voltage will be zero. It is thus immediately apparent that it is possible to adjust the brushes *S* and *S'* in such a way that the current passing through them will increase when there is a decrease in the power factor of the load (an inductive load being assumed); and this will enable an efficient compounding of the generator to be obtained over a fairly wide range of variation of the power factor. Under no-load conditions the converter does not operate, and the voltage is regulated by adjusting the usual field rheostat *R* of the exciter. When running on load, the exciter receives the required super-excitation through the converter and an auxiliary field winding.

The results obtained with this arrangement on a number of machines were extremely satisfactory, and the equipment was then redesigned so as to combine the exciter and the converter into a single unit, (Fig. 2).

The armature is of standard d.c. design with a laminated commutator, but in addition it is provided with two rings connected to electrically opposite points; it corresponds, therefore, to the armature of a single-phase converter.

The field has three windings: the winding *D* is a usual circuit for parallel excitation, and the windings *F* and *G* have functions which will be described later. The commutator has two main brushes *X*, *X'*, and two auxiliary brushes *Y*, *Y'*, with a phase-difference of 90 electrical degrees with regard to the main brushes.

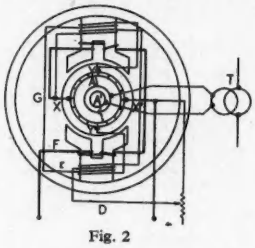



Fig. 2

When the rings *A*, *A'* are not passing any current and the dynamo is running without a load, it will behave as an ordinary machine with parallel excitation; when the main brushes *X*, *X'* are in the neutral zone, there will be no potential difference between the auxiliary brushes *Y*, *Y'* and therefore no current will flow through the excitation circuit *G*.

When the dynamo is generating current, the transverse field of the reaction will produce, between the auxiliary brushes *Y* and *Y'*, a potential difference which is detrimental for the purpose under consideration, and therefore the field is provided with a compensating winding *F*, in series with the supply circuit of the dynamo; this circuit can be adjusted so as to maintain a zero voltage between *Y* and *Y'* no matter what current is being generated.

If, under these conditions, an alternating voltage is applied across the slip rings *A* and *A'*, with a frequency corresponding to the speed of rotation of the machine, and its phase is suitably adjusted, a steady d.c. voltage will be generated at the brushes *Y* and *Y'*, and its amplitude will depend on the current passing between *A* and *A'*, and on the phase angle. The current which will then flow through the excitation circuit *G* will contribute to the total excitation of the dynamo, and determine the variation of the current delivered.

This machine can, therefore, be used to obtain the super-excitation, under load, of a mechanically coupled alternator, the main requirement being that it should be designed with the same number of poles as the alternator, and adjusted to the correct phase angle relatively to the latter. The connection between the slip-rings *A*, *A'*, and the alternator circuit is obtained via a transformer *T* of suitable characteristics, which can be determined by the usual methods, with the primary in series with the consumer line.

It should be noted that the compensating circuit *F* serves only to cancel out the e.m.f. induced between *Y* and *Y'* by the transverse flux of the reaction. As this result can be obtained without particular consideration of the pattern of the resulting field, this circuit can be realized in a simple manner by means of two coils which are oppositely wound on the two projections of a pole with a central groove, as shown in Fig. 2.

Figs. 1 and 2 refer to a single-phase alternator, but the principle can, of course, also be applied to three-phase alternators.

The results obtained were in agreement with expectations, but there is no appreciable saving in the construction of the combined unit, and the simpler arrangement shown in Fig. 1 is therefore just as appropriate for obtaining a constant voltage.

*See Italian Patent No. 400, 729, July 22, 1946. Another exciter of a different type but designed for the same purpose was described by K. Tadel in "E.T.Z.", April, 1946, p. 442.

# DESIGN AND WORKSHOP

*A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.*

## Die-Setting and Economical Punch Press Operation

By E. H. GIRARDOT. (From *The Iron Age, U.S.A.*, Vol. 162, No. 21, November 18, 1948, pp. 94-98, 6 illustrations.)

THE importance of proper set-up of punch press tools is frequently underestimated. Extra costs and other losses which are actually caused by improper die set-up are often erroneously charged to faulty die design or construction, because the troubles experienced cannot always be clearly traced to the set-up. While broken or chipped die sections may be caused by faulty design, construction, or the use of tool steels not suited for the job at hand, an analysis of the die setting methods used should be made before such conclusions are reached.

A typical press shop which produces a varied general class of punched parts carries a heavy investment in machinery. In addition, considerable continued expense is involved in keeping the presses in good operating condition.

Although it would be difficult to establish definite figures representing normal maintenance costs in any shop, a careful examination of actual expenditures shows many items traceable directly to press overloads resulting from improper setting of the dies. Reduced maintenance expense can thus be anticipated through the inauguration of a regular training procedure for die setters.

Forming, bending, embossing and other die types, wherein the work is bottomed, require considerable experience on the part of the die setter to ensure proper set-up. Tests made in hydraulic presses to determine the pressures required by various dies which bottom indicate that little appreciable effect is noted in the punching after overloads as great as 100 per cent if the punching is of sufficient area to resist coining.

For example, a formed part was satisfactorily produced with a pressure of 30 tons. Even when this pressure was increased to 60 tons, the punchings produced at each pressure could be clearly identified from each other.

The set-up of such dies tests the skill of the die setter, since the common screw-adjusting pitman must be set by feel. All presses are so designed that the deflection of their members will remain well within the elastic limit of the materials of which they are made, when exerting full rated pressure. This is shown in Fig. 1. If the ram is set too deeply, the press frame and bolster would be deflected to accommodate this excessive depth setting, resulting in press overload, as shown in Fig. 2.

In this event, the frame would assume a permanent set, resulting in misalignment of the ram and bed faces. This misalignment might assume either of two forms, or a combination of both.

If both sides of the press frame were to stretch evenly, an out-of-parallel condition of the ram and bed faces from front to back of the press would result. If only one side of the frame were stretched, the out-of-parallel condition would be from right to left. An uneven sketch of the frame sides would induce a combination of these two forms of misalignment.

Another objectionable condition caused by overloading is a misalignment of the ram ways to the press bed, which causes an improper angular motion of the ram in relation to the bed. Since a major overhaul is required to restore a press misaligned because of overloading, it is essential that overloading be avoided in order to keep press maintenance costs at a minimum.

If a misaligned press of this type were used for blanking work, particularly on thin materials, a greatly reduced number of punchings between die grinds would result, because the punch and die members could not be properly aligned. Economical operation of blanking dies can be attained only if the cutting faces of the punch and die members are maintained parallel to each other during the cutting cycle, and if the clear-

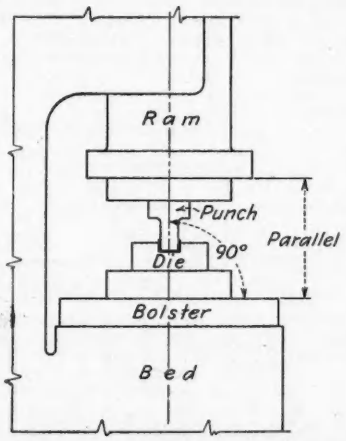



Fig. 1. **Theoretical relationship of ram motion, ram and bolster faces at bottom of work stroke.**

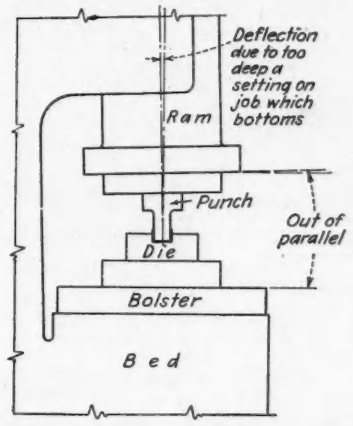



Fig. 2. **Too deep a ram setting on jobs that bottom cause press frame deflection.**

ance provided between the punch and die be evenly divided around the contour.

Some dies may have shear provided on either the punch or die to reduce the cutting pressure required In this event, it is still essential that the die clearance be evenly divided around the blank contour to avoid injurious metal-to-metal contact between the punch and die members.

Parallelism of punch and die members and an even distribution of the die clearance cannot be attained if an out-of-parallel condition exists in the punch press. A die clearance of about 6 per cent of the material thickness is commonly used, so a clearance of only 0·0015-in is usually provided at any point between the punch and die when cutting 0·025-in thick material. It is evident that only a slight misalignment would result in metal-to-metal contact, with resultant injury to the cutting edges.

Draw dies for round shells which are run in single-action presses equipped with cushions require special set-up knowledge. Blankholding pressure must be evenly distributed over the blank area and of sufficient force to prevent wrinkling without retarding the natural flow of material into the die. This optimum pressure can only be determined by trial, with reliance on the experience and good judgment of the die setter.

Adjustment of blankholding pressure is a relatively simple matter with pneumatic or hydraulic cushions, but becomes more difficult with spring cushions. In the latter case, pressure is varied by the amount of initial compression given to the springs by manual adjustment.

Draw, or blank and draw dies that are run in double-action presses require the same critical attention to proper blankholding pressures. Here, however, adjustment is made by means of studs located in the four corners of the blankholder ram. Obviously, considerable skill is necessary to ensure an even distribution of force over the blank area. Regardless of whether a single or double-action press is used, the application of either excessive or insufficient blankholding pressure results in an abnormally high percentage of rejected work.

The set-up of draw dies in double-action presses is another situation that is often mishandled. The draw punch must enter the die centrally, so that the clearance is evenly distributed about the contour. The use of dummy shells or strips of material, inserted around the contour between punch and die, is not recommended since no assurance is given that the punch ram is not crowded to one side in its ways.

A more satisfactory method is to mount the draw punch securely in its ram, then lower it into the draw die and use feelers to divide the clearance evenly before clamping the die. In this manner, centralized location is ensured, since the punch ram is hanging free in its ways and the die is located to suit the punch.

Draw work on other than round shapes may impose difficult blankholding problems. It is sometimes necessary to vary the pressures over the blankholding area and to apply greater pressure at certain points. Such jobs further complicate the set-up problem and require maximum experience and the use of extremely good judgment on the part of the die setter.

Die life and maintenance, major items in press operation, may be directly affected by careless or improper set-up. On dies which bottom, such as form dies, excessive pressures tend to coin the punch and die sections into the softer material of the die set, with subsequent misalignment of the punch and die members.

In the case of cutting dies, improper alignment of the punch and die members in set-up may seriously affect die life and maintenance costs. For example, since a die clearance of 6 per cent of the material thickness is recommended, there is a clearance of only 0·0009-in between the punch and die when cutting 0·015-in stock. This means that almost perfect alignment of the die members must be established during set-up, and maintained when clamping them to the ram and bolster.

Extreme caution should be exercised to ensure that the upper holder is securely clamped flat to the ram face before the lower holder is fastened to the bolster. This is particularly true in the case of punch holders held to the ram by means of shanks. Dirt or burrs on the shank, the top surface of the punch holder, the ram face, or in the ram shank hole may all keep the punch holder slightly out of alignment when the die is set up. Under cutting pressure, the punch holder may be forced into proper relation with the ram, causing misalignment with the die. Very little movement is required to result in metal-to-metal contact between the punch and die, with a resultant decrease in the number of punchings produced between grinds.

The die setter is in an excellent position to contribute towards the design of better and more trouble-free dies. He is aware of operating troubles that occur under production conditions and should call these to attention, so that they can be corrected.

Safety is necessarily a prime consideration in punch press work, and the die setter holds a prominent position as a key safety man. It is his responsibility to mount proper guards to provide maximum protection with a minimum of interference of the operator. Primary instruction in safety practices and operating methods can best come from him to the press men.

Most shops have no established routine for the training of personnel in die setting procedures. Quite frequently, press operators who have set up dies for themselves are designated as die setters. Their experience then becomes a matter of picking up sufficient information to enable them to carry on their work. This information is usually obtained from other die setters who also have had little or no special training in this work.

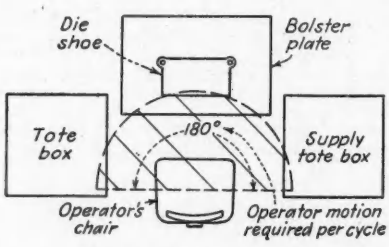



Fig. 3. **This is an inefficient work place set-up for small work. A 90° motion of each hand is required to feed manually and remove the punching from the die.**

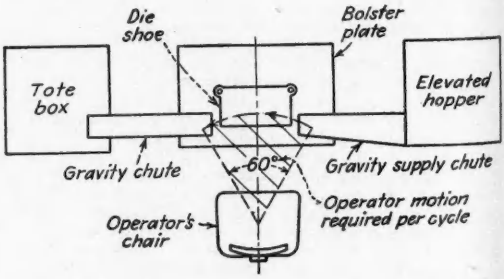



Fig. 4. **In this work place set-up for small work, only a 30° motion of each hand is required. Parts are carried to the feeding station by gravity chute from a hopper, or by conveyor. Drop delivery may be effected by gravity chute.**

An organized program for instructing personnel in die setting provides the best means of safeguarding against losses due to improper set-up. Such a program should include both instruction and practice on a number of points, including the following :—

1. *General Theory of Presses and Dies*

Principles of every type of press and die used in the particular shop should be covered, with emphasis on both proper and improper conditions. For example, if the die setter understood the purpose and function of die clearance on cutting dies, he would have more incentive for being careful in his set-ups to ascertain that this clearance was evenly divided around the die contour. Again, if he understood that excessive pressure on a bottoming type of die contributed nothing to the accuracy of the punching, but would damage the press and/or die, he would be more on the alert to avoid this condition in his set-ups.

2. *Clamping Theory*

Include proper and improper clamping procedures. Punch holders held to the press ram by shanks only must be pressed tightly against the ram face and securely fastened before clamping the lower member. Those bolted to the ram should have these bolts securely tightened, while the lower die member is free to adjust itself to any movement of the punch holder during the bolting process. The lower member can then be clamped.

3. *Set-Up of Press Attachments*

In setting up mechanical feeds, use and adjustment of material guides, adjustment of material movement, release of the material by the feed fingers or rolls to allow final and accurate positioning by the pilots provided in the die should be understood. The principles of air, hydraulic and spring cushions should be studied. Caution must be exercised that all pins between the cushion and the die pressure plate should be of equal length, and that the proper method of adjusting pressure of each type of cushion should be known.

4. *Set-Up of Press Work Area*

In order to obtain maximum efficiency on certain types of jobs, special consideration must be given to the set-up of the work place area. The placing of the material supply and the manner in which the material is removed from the die become important.

The operator's motions should be limited, as far as possible, to the simpler man-motions, such as those involving finger, wrist, lower and upper arm. The use of properly located supply racks and conveyors is important. The operator should not be required to bend over or turn around to reach for a new part or to put down the completed part, if it can be avoided. Such an inefficient work place set-up is shown in Fig. 3. When a satisfactory work area set-up has been made for a particular job, the die setter must be able to duplicate it whenever the same or a similar job is set up in the future. A typical satisfactory work place set-up is shown in Fig. 4.

5. *Safety Practices*

Since the die setter can be considered as a key safety man, he should be given intensive and repeated instruction in safety practices. Careless and hurried set-ups should not be tolerated in modern punch press practice, because they are responsible for a large portion of press and die maintenance. An adequate training program, planned to suit each individual shop condition, will readily pay dividends in the form of more economical punch press operation.

# Friction-Sawing Facts and Figures

By H. J. CHAMBERLAND, Consulting Engineer, Springfield, Massachusetts, U.S.A.

WITHIN recent years, considerable research in the field of friction-cutting by the band saw method has been done by the various manufacturers of metal-cutting saws. Fortunately, results have exceeded expectations, but much remains to be accomplished. It is well to bear in mind, however, that all claims of efficiency at the present time have originated in the laboratory in the form of consumer tests; and for this reason it is, therefore, advisable in the course of general friction-sawing practice to keep some distance from the boundary line of maximum material thickness limitations, rather than attempt to overstep it.

It seems only a short time ago that friction-sawing 20 lineal inches of $\frac{1}{2}$-in armour plate per minute was considered an astounding feat; now, a $\frac{3}{4}$-in thickness and a 13 lineal inches per minute cut is regular routine. Assuming that $\frac{3}{4}$-in is a fair friction-sawing height, this by itself affords ample opportunity to take advantage of the brisk revival of this old process, since cutting rates increase rapidly as thickness of cut decreases.

The truth is that a $\frac{3}{4}$-in, or even a 1-in thickness, is by no means the limit to-day, providing specially designed equipment is used and certain rules are attentively followed. On the other hand, there always exists the tendency for all of us to be over-confident when some new process shows signs of unusual posibilities. With this thought in mind, technicians recommend that even experienced friction-sawing operators regard 1-in as a limit, unless advised otherwise by manufacturers of the equipment. As for the operator who is just feeling his way in friction-sawing practice, he is well advised to adhere to the $\frac{1}{2}$-in limit. Such an application is shown in Fig. 1.

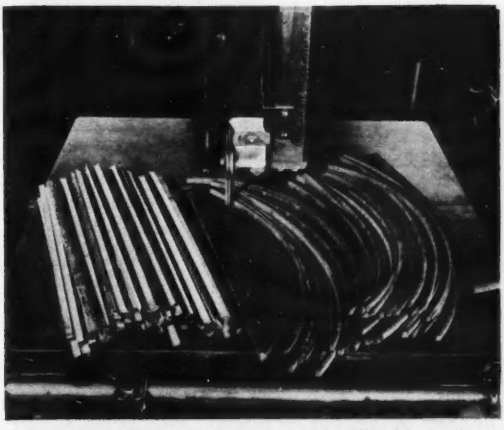


**Fig. 1. Straight and contour cuts produced on $\frac{1}{2}$ in. armour plate with saw velocity of 13,000 fpm. Straight cutting rates on this tough material are from 18 to 20 lineal inches per minute on the $\frac{1}{2}$ in. height. Even the minimum contour sawing rate ranging from 8 to 10 lineal inches per minute is at least 5 times that possible when actually sawing at 100 fpm.**

Sections over $\frac{1}{2}$-in thick should be cut by the "rocking" work feed technique. With friction-sawing, the hardness of the material is of no consequence, since, as in Fig. 2, high-speed milling cutters are salvaged by this process; but the melting point of the material is

very important. All materials within the low melting point range are in the friction-sawing category. In other words, any metal lending itself to flame-cutting likewise lends itself to friction-sawing.

Heat penetration into the side wall of the finish cut is almost negligible. While it increases in depth with increased material thickness, it never exceeds ·002-in. However, heat penetration due to friction-sawing is inherently less than that produced by flame-cutting and, for this reason, sections of a substantial thickness which would usually be flame-cut are often friction-sawed to minimize this defect. In general, only two effects of heat penetration can be found when friction-sawing ferrous metals that respond to heat treatment.

In the case of untreated metal responding to air-hardening techniques, the material immediately adjacent to the cut will be found to have an extremely thin skin hardness due to the temperature reached by friction. So long as the material responds to air quenching, the air entrained in the rapidly moving saw

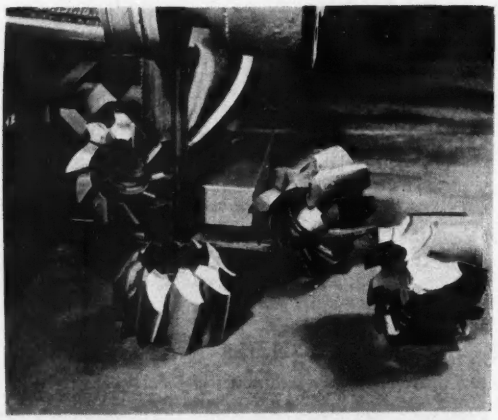


Fig. 2. **Salvaging high-speed steel side mills by friction-sawing at 15,000 fpm. Actual cutting time is 23 seconds on the 2½ in. diameter. Were this a solid cut it would, of course, be outside material thickness limitations of the process.**

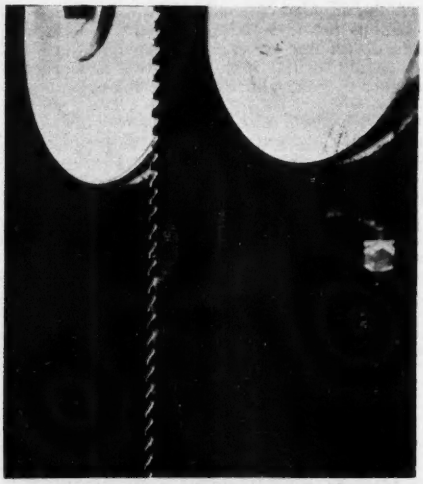


Fig. 3. **A new friction-sawing blade will not reach its peak of efficiency, or generate the required frictional heat, until the teeth appear as shown herewith. After some 20 minutes of cutting, the band will maintain top performance so long as there are signs of teeth.**

The pitch depends on material thickness, the 18-pitch saw applies particularly to thin sheet steel. Widely spaced teeth have a tendency to remove material faster than it can be softened, while teeth too close together will readily load up.

Of the utmost significance is the realization that a metal-cutting saw is not definitely a friction-sawing tool unless heat-treated so that depth of tooth hardness extends to and not beyond the gullet line with absolute control over uniformity. A saw with teeth hardened in this manner will withstand the strains of friction sawing for many hours of continuous cutting.

Why the excessive heat generated has no effect on a high-grade saw band is no phenomenon. The blade is some 255-in long and revolves over 36-in wheels. It is true its velocity often attains almost three miles per minute, but the section doing the cutting is in

cools the material rapidly enough to cause minute skin hardness.

Behind the zone of hardness will be found a very narrow annealed area so small that it is not objectionable. Friction-sawing, therefore, does an excellent job in cases where heat distortion due to machining presents a problem. In the case of heat treated materials which do not react to air quenching, only a very narrow zone of annealing, as previously mentioned, will be found.

The correct type blading as well as velocity is a controlling factor in friction-sawing economy. A rake set saw ("A" temper and 10, 14, or 18 pitch) is most effective. Tooth sharpness is immaterial, because a new saw reaches its peak of cutting efficiency only after it has been in action for some 20 minutes, and the teeth appear as in Fig. 3.

| Material | Thickness, inches | Saw Pitch | Velocity fpm | Cutting Rate per minute |
|---|---|---|---|---|
| A—Manganese Steel | 5/16 | 14 | 5000 | 9 lineal inches |
| B—Nickel Steel | 1/4 | 10 | 9000 | 70 " " |
| D—Non-Magnetic Steel | 1/4 | 10 | 5000 | 50 " " |
| Stainless Steel | 3/4 | 10 | 12000 | 17 " " |
| High-Alloy Steel Casting Gate Cross Section | 1 | 10 | 13000 | 25 seconds per cut |
| Stainless Steel 18-8 | 3/32 | 14 | 6000 | 70 lineal inches |
| Stainless Steel | 1/8 | 14 | 4000 | 32 " " |
| Stainless Steel | 3/8 | 10 | 12000 | 6 square inches |
| Stainless Steel | 1/32 | 18 | 4000 | 400 lineal inches |
| High-Carbon Steel | 1/8 | 10 | 8000 | 27 " " |
| Molybdenum Steel | 1/4 | 10 | 10000 | 31 " " |
| High-Carbon Steel | 1/4 | 10 | 10000 | 80 " " |
| High-Speed Steel | 7/16 | 10 | 12000 | 20 " " |
| Saw Stock | 3/16 | 14 | 5000 | 22 " " |
| Non-Magnetic Steel | 1/2 | 10 | 7000 | 40 " " |
| High-Carbon Steel | 1 | 10 | 14000 | 3¼ " " |
| Sheet Steel | 18 gauge | 14 | 8000 | 156 " " |
| Lead-Coated Steel | 20 gauge | 14 | 6000 | 633 " " |

TUBING

| Diameter | Wall Thickness | Cutting Rate |
|---|---|---|
| 4½ in | 1/4 in | 1·05 mins. per cut |
| 2 7/16 in | 3/16 in | ·04 mins. per cut |
| 4½ in square | 18 gauge | 6 secs. per cut |
| 1½ in | 3/16 in | ·07 mins. per cut |

contact with the material for only a fraction of a second, so that the small amount of heat generated is readily dissipated as the band revolves.

The nature of the material to be cut and its thickness governs saw velocity. Various alloys added to iron change the softening characteristics of the steel. This, obviously, makes it necessary to change the amount of heat to be generated at the point of material saw contact with additional heat being required as material thickness increases. Heat requirements are, therefore, met by adjustment of the saw velocity, the speed range for general friction sawing varies from 3,000 to 15,000 fpm. In actual practice, 12,000 fpm should be accepted as the maximum. Highly experienced operators do, however, slice through the thicker sections at as much as 15,000 fpm.

When friction-cutting, it will be noticed that there is considerable resistance as the work first contacts the saw, but the cutting action starts almost immediately at the moment work pressure and high saw speeds create heat. Uniform work pressure is most essential all through the cutting period. The thicker the section, the heavier the feed required. When straight cutting, the blade should be 1-in wide to withstand the edge thrust caused by heavy feeding pressure. When a curve or radius is involved, bands as narrow as 1/4-in may be used, but the cutting rate will, of course, be reduced accordingly.

The recently introduced "rocking" method of handling the work in the course of the cutting process now makes it possible to increase cutting rate on the heavier sections. As previously mentioned, the thinner the section, the greater the cutting rate. The new technique automatically provides means to keep the heat concentrated and thereby accelerate the cutting action. The principle lies in raising the material from the machine table, a primary step to present a continuous thin section to the saw. This is accomplished by slightly, but uninterruptedly, rocking the piece and meanwhile maintaining the required and likewise constant feeding pressure. It is recommended that the operator should start with a $\frac{1}{2}$-in thickness, but he will soon find himself cutting $\frac{3}{4}$-in and even 1-in steel.

The friction-sawing process as applied to tubing is proving as profitable as with flat stock, but also in this case we cannot overlook certain limitations.

When sawing tubing or pipe, maximum wall thickness is approached momentarily, and is reached only when the saw breaks through the inner wall. Due to the cross-sectional variation, one may readily understand why friction-sawing in this respect must necessarily be limited to a certain wall thickness per given diameter. The following figures indicate maximum wall thickness per given diameter of tubing.

| *Diameter* | *Wall Thickness* |
|---|---|
| 1 in | Solid Round |
| $1^1/_4$ to $1^1/_2$ in | $^1/_4$ in |
| $1^3/_4$ in | $^3/_{16}$ in |
| 2 in | $^5/_{32}$ in |
| $2^1/_4$ | $^1/_8$ in |
| $2^1/_2$ in | $^7/_{64}$ in |
| $2^3/_4$ to $3^1/_4$ in | $^3/_{32}$ in |
| $3^1/_2$ to $4^1/_2$ in | $^5/_{64}$ in |
| $4^3/_4$ to $6^1/_4$ in | $^1/_{16}$ in |
| $6^1/_2$ to $8^1/_4$ in | $^3/_{64}$ in |
| $8^1/_2$ to 12 in | $^1/_{32}$ in |

Velocities for friction-sawing tubing range from 6,000 to 10,000 fpm with 8,000 fpm being most frequently used.

Production data given on the opposite page with saw control factors which made them possible, are but a selection from hundreds of industrial friction cutting applications by the band saw process. Readers may, from this list, note representative results of a nature to be applied to their individual problems.

# A Conical Boring Head

By S. N. FILONENKO. (From *Stanki i Instrument, Russia*, No. 4, 1948, pp. 22-23, 3 illustrations).

THE conical boring head was originally designed for machining the working surface of a compressor cylinder block and it permits of carrying out the rough-turning and finishing cut in one operation. The diameter of the cylinder in its rough state was 61 to 62 mm and it had to be bored to a final diameter of 75 mm. The angle $\alpha$ of the cone is determined by the thickness of material to be removed, see Fig. 1.

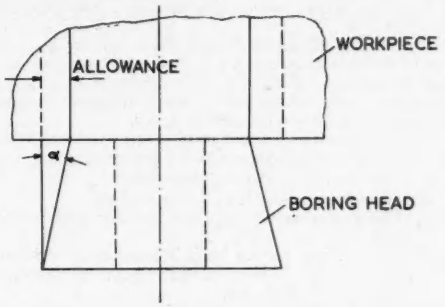



Fig. 1.

The boring head (Fig. 2) has milled slots for the cutting knives, the slots being at an angle to the centre line so as to obtain the correct cutting angle. The angle of the cone being comparatively small (10° 26′), heavy radial forces are obtained. The knives, 3, 4, 5 and 6, are manufactured by milling, and hard cutting edges are welded to the knives. The knives themselves are held in the boring head by means of locating

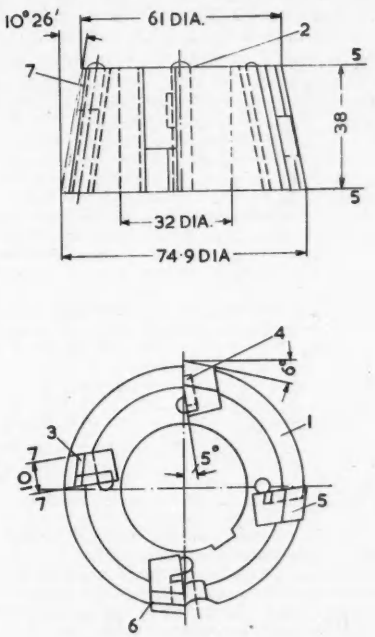



Fig. 2.
1—Boring head, 2—pin; 3, 4, 5, 6,—knives; 7—cutting edge of tungsten steel.

pins, while the cutting edges are located so as to overlap one another by 2 mm.

The head was tested on a horizontal boring machine and entirely satisfactory results were obtained. Paraffin was used as cooling medium and the surface finish was found to be comparable with that obtained by reaming; no vibration was experienced, in spite of the fact that the cylinder block had a port. After machining four cylinder blocks, the knives did not require regrinding.

The thickness of the chip can be determined from

$$a = s \sin \alpha$$

where $s$ represents the feed in mm/revolution and $\alpha$ the angle of the cone. The thickness was calculated as 0·083 mm.

For larger diameters of cylinders, various improvements to the design of the boring heads can be introduced, such as making the knives adjustable, attaching a rough cutting tool in front of the boring head in order to remove the skin of the cast iron, etc.

Compared with the more common cylindrical boring head, the conical head has many advantages:

(1) The machined surface is cleaner.
(2) All cutting edges are equally loaded.
(3) The thickness of chip is equal for each cutting tool.
(4) The loading of each knife is lower as the thickness of the chip is very small; wear is, therefore, considerably reduced.
(5) The conical boring head increases output considerably and does away with the necessity of reaming the surfaces.

## Special Electrodes for Depositing Cutting Edges on Cutting Tools

By V. V. Danilevsky. (From *Avtogennoye Delo, Russia*, No. 2, 1948, pp. 26-27, 1 illustration.)

This method has been developed in order to save high-speed cutting steel. It has been found that cutting tools manufactured by this method equal in quality cutting tools made entirely of high-speed steel.

To obtain the required properties, the electrodes receive a special coating. After manufacture, the cutting tool is subjected to thermal treatment and, subsequently, to the standard mechanical treatment.

In view of the requirements of resistance to wear and to high temperatures, the electrode coating must be such as to give a composition of metal deposited by welding as in Table I.

Table I. Composition of Deposited Metal in per cent.

| Carbon | Molybdenum | Chromium | Tungsten | Vanadium | Nickel | Manganese | Silicon | Phosphorus | Sulphur |
|---|---|---|---|---|---|---|---|---|---|
| 1·0–1·15 | 2·5–3·5 | 4–5 | 5–6 | 2–3 | 0·35 | 0·45 | 0·4 | 0·03 | 0·03 |

To obtain this composition of the deposited metal the composition of the coating should be as follows:—

Chromium (from ferro-chromium), molybdenum (from ferro-molybdenum), vanadium (from ferro-vanadium), tungsten (from ferro-tungsten), ferro-silicon, ferro-manganese, ferro-titanium, chalk, marble, quartz, fluorspar.

Each component must be ground very fine, the ferro-alloys to pass through a 2500/cm² mesh, the remainder through a 1600/cm² mesh. The components are then mixed thoroughly and soluble glass is added at the rate of 17 grams to 100 grams of dry mixture. The wet mixture must pass through a 400/cm² mesh.

The electrode core consists of a low-carbon steel wire of 5 mm diameter. The wire is cut into lengths of 450 mm and two coats are deposited on it. After each coat, the electrodes are dried for 1½ to 2 hours at room temperature. The electrode diameter, after the second coat, should be 8 mm. After drying at room temperature is completed, the electrodes are dried in an electric furnace at a temperature of 150-200° C for 40-60 minutes.

The weight of the coating of the finished electrode must not exceed 37-40 per cent of its total weight.

The metal forming the cutting edge is deposited in a copper mould on the prepared shank of the tool, which is made of a high-carbon steel. The electric arc requires a current of 200-250 amps. When the deposited metal is sufficiently solid, the tool is removed from the mould and forged. Before forging, the tool is heated to 1100° C and forging must be completed at a minimum temperature of 850° C. The tool is then annealed in the normal way.

## Electroplating Recessed Areas with Dipolar Electrodes

By U. C. Jones. (From *The Iron Age, U.S.A.*, Vol. 162, No. 15, October 7, 1948, pp. 98-99, 2 illustrations.)

Recesses or angles in parts to be electro-plated have been a major problem for as long as electro-deposition has been practised, and the development of automatic techniques has accentuated difficulties in many instances by necessitating additional anode rails, rack contacts or contact fingers all of which often prove costly or clumsy or both.

Frequently such problems can be economically solved by the use of dipolar or auxiliary electrodes having no direct electric connection with either the cathode or anode of the bath. Such electrodes pick up stray current and, by affording a path of lesser resistance, can be utilized to throw the specified plate thickness to the recessed area without wasteful heavy plating on the forward sections.

The principle of the dipolar electrode is to provide a short-cut for current in the bath and to lead it into the recessed area where that end of the electrode, being near the cathode, becomes an induced anode and throws metal to the part being plated. The collecting plate or target adjacent to the anode becomes an induced cathode and is plated upon. The other end of the dipolar electrode is positioned for optimum plating of the recess and becomes an induced anode.

Current, rather than going through the solution, is drawn to the collector plate, flows down the metal conductor to the induced anode, shaped to conform to the recess. The size of the collector plate and its proximity to the anode determine the metal distribution on the part. In turn, resistance of the solution, current density, distance of the plate from the anode and other factors determine the size of the collector plate.

The principle is in operation when plating on tank walls occurs. If uninsulated steel tanks are used in zinc, silver or cyanide copper plating, and the distances of the racked work from the anode are not correct, metal is deposited on the tank wall and the tank near the work acts as an induced anode. Or, in other processes when there is a break in the tank lining, a similar phenomenon occurs, and burned or otherwise defective work as well as metal depletion from the bath result.

# Special Light Alloys for Electro-Polishing

By J. HÉRENGUEL and R. SEGOND. (From *Revue de l'Aluminium, France*, Vol. 25, No. 148, October, 1948, pp. 306-310, 6 illustrations.)

THE industrial potentialities of electrolytic polishing have developed rapidly for about 10 years, and now command lively interest. However, many potential users are confused by the diversity of the electrolytic brightening and polishing processes available. This is made worse by the varying degree of success that has actually been achieved. The mistaken belief quite often prevails that these processes compare with mechanical polishing to the extent that an evenly polished surface can be produced on any article, regardless of the metal of which it is composed.

The factors governing the choice between the available processes may be roughly grouped as follows:

(1) Use of product and required quality of finish.
(2) Degree of roughness of the untreated surfaces.
(3) Properties of the material to be treated.

This article does not present any contribution to knowledge of factors in the first group. It is, however, important at this point to stress the factors of the second group. While mechanical polishing seems capable of levelling surface projections without removal of an equivalent amount of metal, electro-polishing proceeds by dissolving them and removing part of the surface as well.

The following conclusions may be drawn:

(a) The untreated surfaces must be free from deep scratches or surface cracks, such as are still too often found after pressing and other operations.

(b) If a high order of finish and of geometrical surface perfection is required, a bath of low dissolving power has to be chosen, and there will have to be some preparatory mechanical polishing.

(c) If, on the other hand, it is desired to use electrolytic processes for a good rough polish, to be followed, if necessary, by a quick and cheap mechanical finishing operation, a high dissolving power is required. Some baths of high dissolving power are sufficiently selective to give a satisfactory finish, such as the sulphuric-phosphoric acid baths described in French Pat. No. 798721 and others.

The remainder of this paper is concerned with the third group of factors, arising out of the properties of the materials used.

## CHOICE OF ALUMINIUM ALLOYS FOR ELECTRO-POLISHING

Leaving aside for the moment the difficulties arising out of surface defects of either a mechanical or geometrical nature, the first problem encountered is that of the possible existence of constituents of a phase distinct from the base metal. The difference often found between the rates of dissolution of these constituents causes disturbances during the electrolytic treatment. It is, therefore, natural to direct our attention to metals of high purity; in this case, the metal in question is the purest available aluminium, that is, 99·9 per cent or 99·99 per cent if possible. This was, in fact, used for electro-polishing soon after its commercial production had been initiated by Gadeau in France about 1932.

However, while aluminium of high purity is suitable for electro-polishing, its mechanical properties are no more than mediocre.

Solid solutions suggest themselves on account of their greater mechanical strength, while they are also electrolytically homogeneous. Magnesium had already been used by H. Schick and W. Helling[1] for anodic oxidation, and for electro-polishing followed by oxidation it would appear even more interesting.

However, apart from the purity of both the aluminium and magnesium, the alloys require certain metallurgical conditions in their preparation and use, to achieve complete success.

All these considerations resulted in the invention of two special alloys, called Brillalumag 3 and 5, which have already been described in a paper presented in 1947[2].

*Composition and Mechanical Characteristics of the Alloys Brillalumag 3 and 5.*

| BRILLALUMAG | | 3 | 5 |
|---|---|---|---|
| Magnesium content | | 3% | 5% |
| Annealed | Compression | 14000 psi | 17000 psi |
| | Tension | 26000 psi | 33000 psi |
| | Elongation | 30% | 28% |
| Cold drawn | Compression | 35000 psi | 49000 psi |
| | Tension | 38000 psi | 53000 psi |
| | Elongation | 4% | 3% |

Brillalumag 3 is the more ductile alloy; when annealed, its ability to undergo deformation is exceptional. When hardened, it is strong enough for many applications, while Brillalumag 5 is used where stresses are too high for the 3 per cent alloy.

## CONDITIONS OF PROCEDURE AND USE OF BRILLALUMAG 3 AND 5.

The alloys must be in homogeneous solid solution at the time of polishing and oxidation if these operations are to be successful. The semi-finished product requires a fine-grained, non-oriented structure, so that the final deformation shall be easy and the surface after the deformation shall be free of "goose pimples," which often make the eventual polishing a prolonged and difficult operation. These two, homogeneity and fine-grained texture, are the essential conditions. They must be fulfilled by the maker of the semi-finished product and maintained by the final manufacturer.

### HOMOGENEITY

Both alloys solidify so quickly that the diffusion of magnesium is always incomplete. The ingots are of heterogeneous structure, particles of $Al_3Mg_2$ being embedded in the basic phase. Reheating of the ingots will not usually bring success of its own accord, but a series of mechanical deformations, each followed by annealing, will lead to the desired homogeneity.

The reason for this is that not only is a solid solution of the magnesium essential, but its concentration throughout the ingot must also be uniform. While the metal will acquire a satisfactory polish as soon as $Al_3Mg_2$ disappears, the rates of oxidation of magnesium solutions in aluminium vary strongly with the concentration of the former. At 15 volts, in a 10 per cent sulphuric acid bath at 20° C, a 6 per cent magnesium alloy will oxidize about twice as quickly as an alloy with about 2 per cent magnesium. Further detailed results were published[3].

This selective oxidation is illustrated in Figs. 1 and 2, which show a bar of uniform phase, but apparently insufficiently worked to give uniform concentration. It has been cut perpendicularly to the direction of rolling, highly electro-polished, and heavily oxidized.

In such cases, it was found that the network of $Al_3Mg_2$ disappeared after one heating above the precipitation temperature, but extensive working and

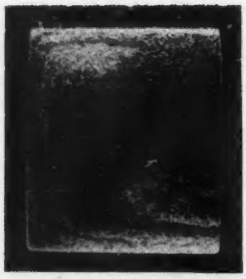


Fig. 1. **Section of a rough rolled bar, of pure Al and 3 per cent Mg, perpendicular to direction of rolling. Full size view after electro-polishing and anodic oxidation.**

annealing were required for good oxidation after polishing.

Precipitation on reheating has to be avoided by annealing at a sufficiently high temperature. This may be done freely with the 3 per cent alloy as the solubility in the base metal is large enough at fairly low temperatures. The 5 per cent alloy requires rapid cooling so that it shall harden properly. The disadvantages arising if precipitation is permitted have been demonstrated[4].

## MAINTENANCE OF A CLOSE-GRAINED STRUCTURE DURING WORKING

It has already been stated that the grain structure should be close, so that the final deformation shall not lead to a granular surface. That it is also important to avoid a marked degree of orientation, which might cause the same defect, has already been proved in the case of pure aluminium[5].

In view of this double condition, it is thought necessary to follow closely the rules which will be enumerated below. It has to be borne in mind that pure metals and their simple alloys are much more prone to crystal growth and preferential orientation than the more common industrial alloys.

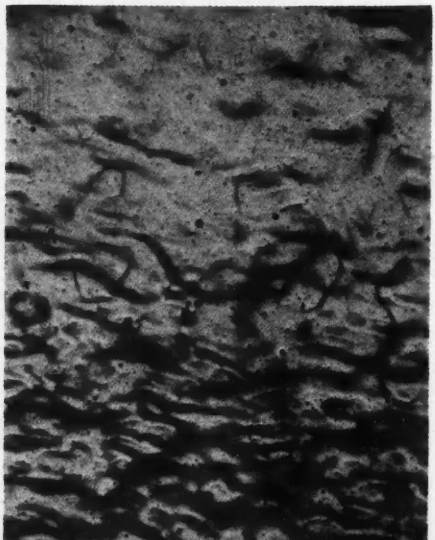


Fig. 2. **Enlargement (× 85) of part of Fig. 1 showing transition from a rough portion to a smoother one.**

Conditions suitable for pure aluminium have already been studied[6]. In the case of the Brillalumag alloys, the following factors have been found to be of importance:

(1) Extent of cold-working after annealing;
(2) Temperature of reheat;
(3) Speed of heating-up.

The time of soaking at the maximum temperature, while it influences grain size, has been found to be a secondary effect only, except when the temperature exceeds the values recommended here.

The elongation during cold-working has the following results. If too small, reheating may result in rapid growth of grain size; if too large, orientation is produced.

For the temperature, the lower limit is that which permits renewed cold-working without orientation, while too high a temperature will cause grain growth.

Quick heating-up was effected in an electric forced-convection furnace of the industrial type for the 3 per cent alloy, while heating in salt baths is preferred for the 5 per cent alloy and enables the highest recommended temperatures to be reached without grain growth. The importance of the rate of heating has also been observed with 99·5 per cent Al[7], and Al–Mg–Si[8].

If these rules are observed, an initial grain size of less than 0·1 mm in the casting can almost be maintained during the working cycles required to obtain homogeneous material. It should, however, be noted that with these materials, it is impossible to rely on obtaining such a fine structure by one final cycle of cold-working plus annealing, once growth has been permitted.

## SPECIFICATION FOR PRODUCTION AND FABRICATION OF BRILLALUMAG

(a) *Casting.*

Composition: Aluminium of 99·99 per cent purity, 3 or 5 per cent magnesium.

Solidification: As quick as possible to obtain close grain and fine distribution of $Al_3 Mg_2$.

(b) *Working.*

This must be intensive enough to obtain such homogeneity as has been described in the semi-finished product. This is best done by cold working and annealing. Hot-rolling of the ingots is to be avoided, as it produces defects which cannot easily be overcome later.

Cold-drawing or rolling after initial extrusion should give minimum elongations of 25 per cent and 200–300 per cent maximum between anneals.

Annealing, with rapid heating in forced-convection furnaces or, better still, in salt baths, should be done above 325° C and below 425° C.

With 5 per cent Mg, it is also necessary to heat above 270° C and cool rapidly to avoid precipitation.

If the alloys are intended for use without deformation, e.g. machining, they should be delivered cold-drawn within the stated limits. If they are intended for pressing, etc., they should be annealed. All concerned, producers as well as users, should note everything pointed out in order to avoid disappointment.

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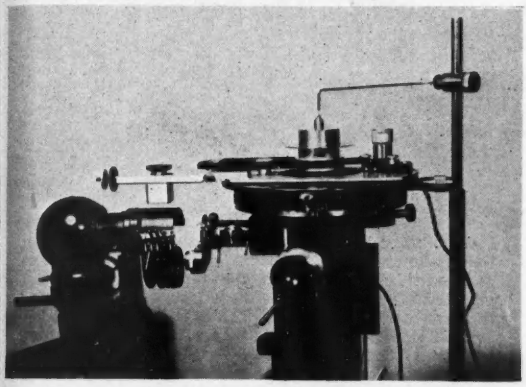
# New Materials, Processes and Equipment

## MICRO-HARDNESS TESTER

Micro-hardness testers of varying design have, in the past few years, been described in scientific publications. A tester designed by E. Bergsman of the Munkfors Works, Uddeholm Company, Sweden, works on a principle different from these in that the specimen and not the indenter is movable, thus simplifying the construction. As the indentation load is applied direct to the specimen, the tester is self-calibrating.



The apparatus has the following characteristics. The lightest load (less than 1 gram) is so small that even very soft, small or thin specimens or small and soft components of a micro-structure can be tested. The heaviest load (500 grams) is great enough to permit impressions to be easily measured even in very hard materials, such as carbides.

The tester can be used with any standard inverted microscope and a Vickers or a Knoop diamond as indenter both for static indentation tests and scratch tests.

## INDUSTRIAL PHOTOCELLS

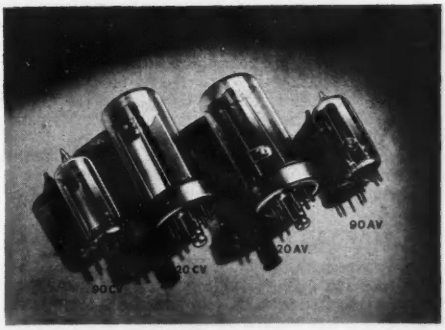
As a result of an extensive programme of research both in material and design, Mullard Electronic Products Ltd., Century House, London, W.C.2, have recently announced a new range of photocells which fulfil special requirements for industrial application. These are: (1) robustness and rigidity; (2) reliability and long life; (3) minimum size with maximum cathode area; (4) maximum permissible cathode current and high sensitivity; (5) single-ended construction with low leakage,; (6) positive location with uniform orientation.

The photocells at present available fall into two main groups, as follows :—

1. Those having a high sensitivity to incandescent and infra-red light, and
2. Those having a particularly high sensitivity to daylight and blue light.

All the photocells in this new range are of all-glass construction, which provides the maximum possible cathode area, at the same time ensuring a high degree of rigidity and freedom from microphony. The design is such that the cells have a reliability and long life under arduous conditions of service. They are available in two sizes, the large " 20 " series on a loctal base and the " 90 " (miniature) series on a button B7G base.

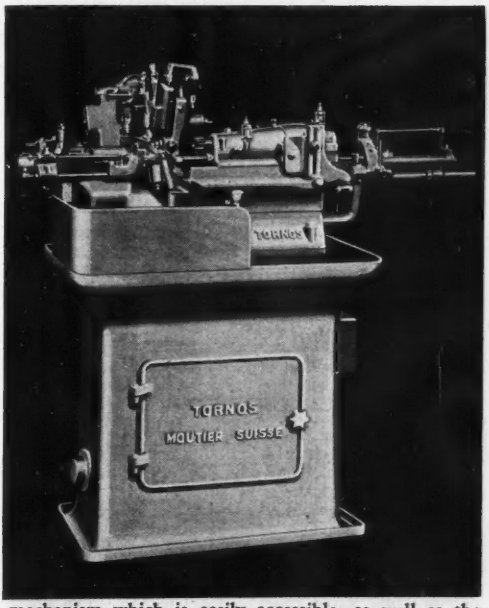
The miniature type give adequate output current




for a great number of applications and are particularly suitable for those equipments in which conservation of space is an important factor. The loctal types have a higher output by reason of their larger cathode area, which renders them particularly suitable for use in standard industrial electronic control equipments.

## WOOD-SCREW AUTOMATIC

The threads of the wood screws produced on this precision automatic manufactured by Tornos Works Ltd., Moutier, Switzerland, are absolutely regular and smooth, perfectly profiled, and ensure easy penetration of wood. The cabinet base contains the driving




mechanism which is easily accessible, as well as the coolant pump and tank. The chucking operation can take place in any position of the headstock, and the thrust of the chucking mechanism is absorbed by the machine bed, thus avoiding undue wear of the headstock slide. The longitudinal feed is governed by a

special mechanism consisting of two bell-crank levers which transform the radial stroke of the flat cam into the longitudinal motion of the headstock. Similar screws of different sizes can be produced without changing the cams.

The machine is fitted with four radial tools, three of which are simple cutters and are used for turning the shank, shaping the head, and cutting off the screw. The fourth tool is a milling cutter, the radial motion of which is controlled by a special adjustable cam. The rotation of the cutter is derived from the headstock arbor. The thread is cut by the generating process, by means of a special device developing the thread at the same time as the headstock advances.

The automatic is equipped with a screw slotting attachment. When finished and cut off, the screws are seized by a feed lever and brought in front of the saw.

## NEW LETTERING INSTRUMENT

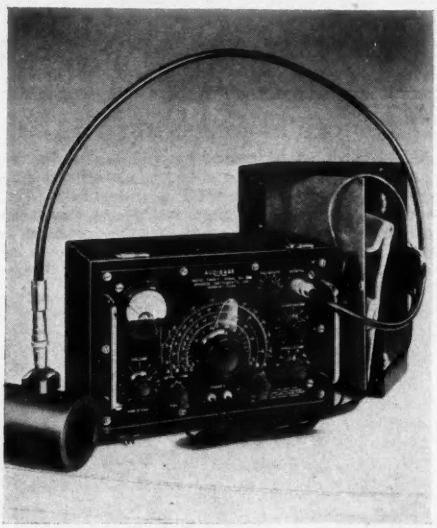
This instrument is a small mechanical device for reproducing many sizes of letters from a single templet directly in ink or pencil on any suitable surface. It is manufactured by Varigraph Company Inc., Lincoln, Nebraska. The letters may be reproduced to any desired width combined with any desired height between 0·075 and 0·750 inch. Letter width and height are controlled by positioning two knobs.

Resting directly on the surface being lettered the Varigraph slides against any straight-edge in being moved from one letter to the position of the next letter. A Varigraph Lettering Templet carried by the instrument slides left and right to position in the instrument the desired letter. To reproduce a letter, a finger operated point follows the letter grooves in the lettering templet.

## THICKNESS DETECTOR

A new model of the "AUDIGAGE" Thickness Detector has recently been announced by Branson Instruments, Inc., Danbury, Conn. It is used for non-destructive thickness measurement of steel, copper, aluminium, glass, unfilled plastics and other materials, from one side. Typical applications include tanks, pipes, ship's hulls, boilers, pressure vessels and sheet metal.

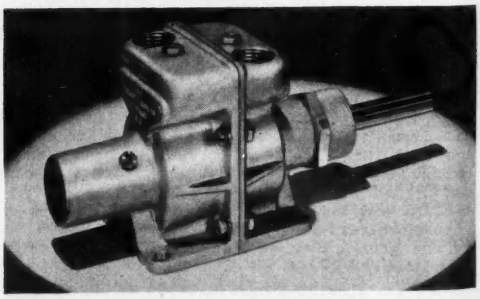
The operating principle of this instrument is as follows: An X-cut quartz crystal in the gauge head, powered by an electronic oscillator, generates ultrasonic waves, which are transmitted into the material under test. The instrument is tuned to a frequency which produces resonance in the unknown thickness. Resonance is indicated simultaneously by a milliammeter and headphones, to provide either visual or audible



signals. The frequency at which resonance is indicated can be converted to thickness by use of a conversion scale on this instrument panel. The thickness reading obtained by this method is the actual wall thickness of the material, directly under the crystal; backup liquids or scale deposits are not measured. While the regular flat crystal gauge head can be used on curved surfaces, a substantial gain in sensitivity is obtained by using special cylindrically curved crystals for pipes of less than 6-in diameter.

## PUMP FOR INDUSTRIAL LIQUID HANDLING

A gearless pump for the industrial handling of liquids, with several special features and improvements, has been announced by Eco Engineering Company, Newark 1, N.J. Chief among the improvements is the new dual eccentric piston design which gives a strong flow against pressure.



Available in four sizes, $\frac{1}{4}$ in, $\frac{3}{8}$ in, $\frac{1}{2}$ in and $\frac{3}{4}$ in, these pumps have capacities ranging from 1 to 12 U.S. gpm varying with pump speed. Pumps are available for use against pressures ranging from zero up to 100 psi. They can be run and will deliver at speeds from 200 to 2400 rpm.

The pumps can be used in various positions and for a wide number of operations. Fitted with self-lubricating bearings, they eliminate problems of lubrication, at the same time giving freedom from fouling of intake or discharge lines by pump greases or oil, obviating contamination of the liquid handled.

# CLASSIFIED ABSTRACTS

*Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.*

## COST ACCOUNTING

### Costs and Profit Calculation at Steel Works

By F. KLING. (From *Jernkonterets Annaler, Sweden*, Vol. 132, No. 8, August, 1948, pp. 268-269, 7 illustrations.)

To begin with, the author pays special attention to some definitions, for instance, the concepts of alternative costs, quantitative and time costs, marginal costs, marginal revenue, and standard costs. Then follows a discussion of problems in connection with cost estimating at a steel works, i.e., evaluation of scrap arising from production, a method for short-cuts in cost estimating of low alloy steel, internal billing, standard cost accounting, and monthly profit and loss statements.

As for internal billing it is stated that the products of a steel works should be divided into certain " natural" groups, such as pigiron, finished hot-rolled, cold-rolled, fabricated and other products, and that the internal billing between these groups should be done at market prices ; billing between stages, ingots—slabs—finished products, should be at actual equivalent costs. By means of such billing as well as standard cost accounting and monthly statements showing the difference between revenue and standard cost, a complete check of sales, production, and the reliability of standard costs is obtained.

This is followed by a description of the method of analysing cost and profits with special reference to marginal cost and marginal revenue.

When selecting orders, calculation should be made not by the ton, but rather by some sort of time unit. The author suggests profit evaluation in relation to the part of total production costs, which is directly proportional to time. This part of the total production costs constitutes the productive preparedness which the company always has to maintain to make it possible at any time to select different products for manufacture. In other words, those orders should be selected that give the largest difference between marginal revenue and marginal costs in relation to that part of total cost of production which is directly proportional to the time factor.

Finally, mention is made of the great importance of the alternative costs when considering orders for small quantities and small sizes. Consideration must be given to setting-up time for the works and to changing of rolls, which have a decisive bearing on costs.

## ELECTRIC TRANSMISSION & DISTRIBUTION

### General Aspects of E.H.V. System Earthing

By H. MEYER. (From *The Brown Boveri Review, Switzerland*, Vol. 35, Nos. 7/8, July/August, 1948, pp. 172-183, 5 illustrations.)

SOLID and resonant earthing are the only two practical methods of operation possible for extra-high voltage power transmission lines, since all other systems only exhibit more drawbacks and no additional advantages.

Solidly earthed systems possess the most favourable voltage conditions, although the short-circuit current may become high. The effectiveness of the earthing and the behaviour of the earthed system depend on the ratio of the zero-sequence impedance to the positive-sequence reactance viewed from the fault location (symmetrical components). The insulation level for solidly earthed e.h.v. systems with earth wire can be reduced to 0·8 of that of resonant-earthed systems. In earthed systems the short-circuit current can, under certain circumstances, become considerably higher for single or two-phase to earth faults than for a three-phase short-circuit, and this must be taken into account in dimensioning the breaker.

Complete neutralization of the earth-fault current in an e.h.v. system with resonant earthing is also theoretically possible when transmitting power over long lines. Its practical application is, however, a question of equipment and, therefore, of cost. Conditions become more favourable, the more the compensating equipment is distributed along the line.

Long-distance power transmission requires consideration of the voltage rise occurring when the line is disconnected, both for solid and resonant-earthed systems. It must be taken into account in co-ordinating the operating voltage of the system and the rated voltage of the equipment. These voltage increases can assume very high values. Methods of diminishing them are indicated.

## HEAT TREATMENT

### Cycle Annealing in Quality Steel Production

By E. THOLANDER. (From *Jernkontorets Annaler, Sweden*, Vol. 132, No. 10, October, 1948, pp. 418-419, 44 illustrations.)

A COMMITTEE of the Technical Research Board of Jerkontoret has studied the problem of utilizing the isothermal transformation of austenite after the hot-working of steel. At first, small samples of four different steels were treated on a laboratory scale in order to investigate the influence of different factors on the results obtainable. These four steels were chosen as representative of a large number of air or oil-hardening alloy steels commonly used in Sweden.

Further investigations were made on billets both on a laboratory and technical production scale. In some cases, billets were used which had been prepared by boring longitudinal or transverse holes and inserting small test pieces in these holes. Thus, a correlation could be made between the results obtained with small samples and those obtained with billets.

For a great number of alloy steels recently standardized in Sweden, data for isothermal transformation to pearlite have been obtained by experiments with small samples, the results of which are given diagrammatically. For all the steels investigated, the temperature range for transformation to pearlite is identical with one or more of four ranges. Only two of these steels could not be transformed one hundred per cent.

Segregation retards the transformation in ingot and large billet section structures, but has no great influence on the transformation in small bar sections. Machinability tests showed good results for pearlitic structures, but bad results for bainitic ones. The hardness of bainitic structures was much higher than that of pearlite.

The conclusion of the investigation is that isothermal transformation to pearlitic structure with good machinability is possible in a short time for most of the common alloy steels of the oil or air-hardening types. Advantages

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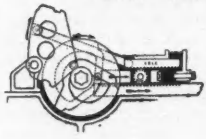
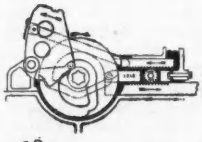
**AUTO POINTS**

Accurate feed strokes for end-working tools are set by simply sliding the setting-block to the appropriate graduation on the scale attached to the quadrant.

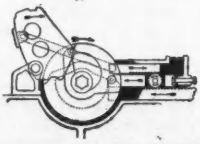
The pentagonal centre-block and two independent slides are actuated by pushers to which the motion of the quadrants is transmitted by racks and pinions, as shown below. Fixed cams, on the same shaft, actuate a fast return and approach motion by means of slides on which the rack pinions are located.



*Slides in back position. Pinion carrier about to advance, giving constant approach stroke.*



*Approach stroke completed. Pinion carrier secured between cams and stop screws.*



*Working stroke completed.*

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*Because there are no cams to change!*

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such as elimination of cracks (but not flakes) by cooling, pickling or grinding, better machinability, elimination of separate annealing operations after cooling, etc., can be obtained by cycle annealing in conjunction with prior hot-working.

## ROLLING MILL PRACTICE

### New Cold Mill at Weirton Increases Tin Plate Capacity

By C. J. KLEIN. (From *Iron and Steel Engineer, U.S.A.*, Vol. 25, No. 11, November, 1948, pp. 37-48, 17 illustrations.)

THE first mile-a-minute strip mill, built for the Weirton Steel Company, required the solution of many engineering problems to take full advantage of the mill. They are explained in a historical review of the company's development from the time it first entered the steel business, in 1906, up to 1947. The details described concern *inter alia* a belt wrapper to facilitate the tedious operation of passing the front end of a strip into a small slot in the reel head on the original tandem mill. A stand 5 was added to the original four stands in 1935 and a stand 6 in 1947. An improved reel head has been developed, which to a large extent, removed difficulties and defects arising from the former design. With the ever increasing capacity of the whole plant, it was necessary to design and produce an improved uncoiler and conveyor as well as a charging and coiling equipment. Each of the roll stands house work rolls 21-in in diameter × 42-in in length mounted in roller bearings, and back-up rolls 53-in in diameter × 41-in in length mounted in tapered neck oil bearings. A new design of tapered end on the work rolls provides for a solid operating mounting of the universal coupling ends, which are readily removable for taking off bearings for roll grinding. The mill drives, including the drive of the reel, consist of 7 motors totalling 17,550 hp. The various requirements of lubrication are met by 6 different systems. Hydraulic water at a pressure of 180 psi as well as mill water at a pressure of 40 psi is available at the operator's discretion. A total of 300 U.S. gpm is used for roll and strip cooling. Special care has been taken to develop a careful system of quality control.

The products are primarily tin mill coils in gauges ranging from 38 (0·0060 in) to 29 (0·0135-in) and in widths up to 38-in, and also sheet mill coils in gauges ranging from the lightest to about 0·025-in and in widths up to 38-in. The coils have an inside diameter of 16½-in and a maximum outside diameter of 72-in. The weights, depending on width, range up to a maximum of 32,000 lbs. The mill has a capacity for rolling 45,000 tons per month.

## SCIENTIFIC INSTRUMENTS

### A Highly Sensitive Differential Manometer

(From *Technical News Bulletin, National Bureau of Standards, U.S.A.*, Vol. 32, No. 11, November, 1948, pp. 132-133, 2 illustrations.)

A DIFFERENTIAL manometer which will detect pressure differences as small as one millionth of an atmosphere over a range of total pressure from 1 mm of mercury to 1 atmosphere has been developed by Harry Matheson and Murray Eden of the National Bureau of Standard's physical chemistry laboratories. This instrument, which operates on the principle of the aneroid barometer, can be used to compare pressures of gases, organic vapours, and non-corrosive liquids. It is particularly useful in vapour pressure work, since less than a drop of liquid is sufficient to compare an unknown vapour pressure as a function of temperature with that of a known liquid. The rest point of the new manometer is insensitive to temperature, and accidental pressure differences as high as an atmosphere introduce only temporary hysteresis effects. The instrument is also readily portable and need not be levelled before operation.

## WELDING

### The Function of the Coating of Welding Rods

By J. D. FAST. (From *Philips Technical Review, Holland*, Vol. 10, No. 4, October, 1948, pp. 114-122, 4 illustrations.)

IN a previous article* the physico-chemical factors which play a part in welding with bare rods have been discussed. In the present article, the requirements that have to be met by the coating of welding rods are analysed. Firstly, the nature of the forces bringing about the transfer of the metal is dealt with and the circumstances under which iron oxide in the coating may cause porosity of the welds are investigated. The protection afforded by the slag against oxygen and nitrogen is discussed, starting from an ideal case where liquid iron is covered with a simple slag consisting solely of liquid $SiO_2$. Attention is then directed to more complicated slags, and, finally, the coatings and slags applied in welding technique are described. It appears that satisfactory protection against oxygen can only be obtained if the use of oxides with a heat of formation less than about 100 kcal per gram atom of oxygen is entirely avoided in the coating. Maximum protection against nitrogen requires the use of slags of low solubility for this gas. In addition to the requirements for protection against oxygen and nitrogen and those for the transfer of metal and the avoidance of porosity, the coating and the slag have to fulfil other functions, which are likewise discussed here. These requirements are so numerous and so divergent that it was necessary to develop various types of welding rods, each possessing a certain number of favourable properties.

*See THE ENGINEERS' DIGEST, "The Part Played by Oxygen and Nitrogen in Arc Welding," November and December, 1948, pp. 375-378 and pp. 413-414 respectively.

## WIRE AND WIRE PRODUCTS

### New Wire Fatigue Testing Method

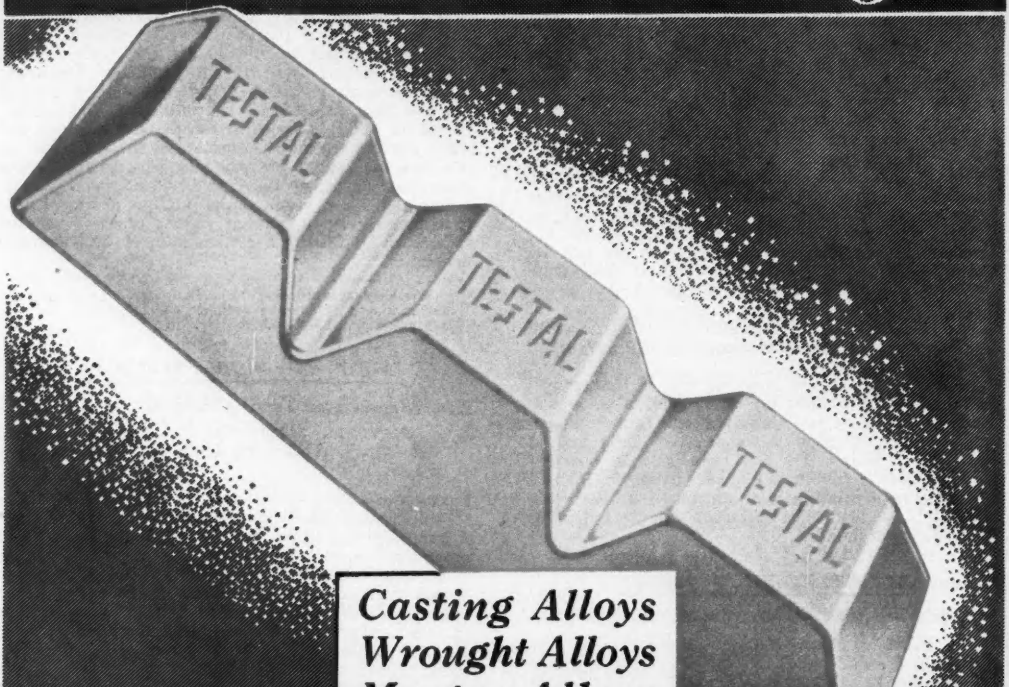
By F. A. VOTTA. (From *The Iron Age, U.S.A.*, Vol. 162, No. 7, August 12, 1948, pp. 78-81, 4 illustrations.)

FATIGUE data are considered to represent the most reliable basis for predicting wire life in dynamic service. A simplified and rapid, yet highly accurate, method of obtaining fatigue strength and endurance limit values to serve as a quality index is described in this article. A new type of instrument, developed some years ago and recently offered for commercial use, is required for the tests, which are carried out in four stages :—(1) Calculating the sample length and chuck-bushing distance 'appropriate to the testing stress desired in the wire to be tested (its modulus of elasticity and diameter being known) ; (2) cutting such a length, clamping it in the chuck, inserting the other end in the proper bushing, and locating the supporting guides ; (3) switching on the motor ; and (4) waiting for the break.

The theory underlying the operation of this instrument is given in this article, together with the results of representative series of fatigue tests. The method is of interest both to the wire manufacturer and to the consumer. It is said to make possible production control as well as material acceptance on the basis of modern statistical quality control.

# TESTAL

## *Aluminium Alloys*

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# NEWS OF THE MONTH

## NEW YEAR HONOURS

Well-known engineers and scientific men figure in the New Year Honours List, published on December 31, 1948. Knighthoods have been conferred upon Mr. A. J. Gill, M.I.E.E., Assistant Director-General (Engineering), General Post Office; Mr. J. H. Goddard, President, of the Machine Tool Trades Association; Dr. H. L. Guy, C.B.E., M.I.C.E., M.I.Mech.E., F.R.S., Chairman of the Mechanical Engineering Research Organisation, Department of Scientific and Industrial Research, and secretary, Institution of Mechanical Engineers; Mr. John Hacking, M.I.E.E., deputy-chairman, British Electricity Authority; Mr. D. MacL. Skiffington, C.B.E., M.I.N.A., J.P., director, John Brown and Company, Limited, Clydebank, and Mr. T. E. B. Young, M.I.Min.E., M.I.C.E., F.G.S., production member of the National Coal Board. Sir Henry T. Tizard, K.C.B., A.F.C., F.R.S., has been created a knight Grand Cross of the Order of the Bath, Colonel P. G. J. Gueterbock, C.B., D.S.O., M.C., immediate past-president of the Institute of Metals, a Knight Commander of the Order of the Bath, and Engineer Rear-Admiral S. O. Frew, C.B., a Knight Commander of the Order of the British Empire.

The new Commanders of the Order of the British Empire Empire include Mr. Atholl Blair, M.B.E., M.I.Mech.E., M.I.N.A., engineer manager and director, Harland and Wolff Limited, Belfast; Mr. O. V. S. Bulleid, M.I.Mech.E., chief mechanical engineer, Railway Executive, Southern Region; Mr. J. R. Caseley, M.I.C.E., deputy director, Opencast Coal Production, Ministry of Fuel and Power; Mr. P. A. Cooke, O.B.E., M.C., director, Engine Production, Ministry of Supply; Colonel W. C. Devereux, F.R.Ae.S., managing director, Almin Limited, and chairman, International Alloys, Limited; Mr. I. G. Evans, assistant secretary, Department of Scientific and Industrial Research; Lt.-Colonel F. G. Hill, M.C., M.I.C.E., deputy chief engineer, Ministry of Health; Mr. V. Kenniff, M.I.C.E., M.I.E. (Aust.), director, Public Works, Hong Kong; Colonel H. L. Lendrum, A.M.I.E.E., Corps of Royal Engineers; Mr. I. D. MacLennan, M.I.E.E., chief electrical engineer, Public Works Department, Sierra Leone; Mr. J. A. Milne, chairman and managing director, J. Samuel White and Company, Limited, Cowes; Mr. James Mitchell, managing director, Iron and Steel Production, Stewarts and Lloyds Limited; Professor F. H. Newman, D.Sc., Ph.D., A.R.C.Sc., F.Inst.P., A.M.I.E.E., University College, Exeter; Captain (E) O. W. Phillips, R.N. (ret.); Rear Admiral (E) H. S. Roome; Alderman Charles Terry, chairman and managing director, Herbert Terry & Sons, Ltd., Mr. Harold Wilmot, President, Locomotive Manufacturers Association of Great Britain; Lt.-Colonel E. H. E. Woodward, M.C., T.D., M.I.E.E., for services to the electricity-supply industry; and Mr. R. T. B. Wynn, M.I.E.E., assistant chief engineer, British Broadcasting Corporation. The list of recipients of the O.B.E. includes the names of Mr. Russell Bailey, M.I.Mech.E., joint managing director, Chas. Roberts and Company, Limited; Alderman George Clark, M.I.Mech.E., J.P., chairman, Chesterfield Local Employment Committee; Mr. T. A. S. Fortune, M.I.C.E., M.Inst.T., general manager and secretary; Leith Dock Commission; Mr. H. C. Hall, M.I.Mech.E., F.R.I.C., metallurical research consultant to Rolls-Royce Limited; G. B. King, joint managing director, Needle Industries, Ltd., Mr. J. A. Lee, M.I.E.E., A.M.I.Mech.E., construction (transmission) engineer, British Electricity Authority; Mr. H. Lithgow, shipyard manager, Swan, Hunter & Wigham Richardson, Ltd., Mr. R. J. Magowan, director, Portadown Foundry, Ltd., Mr. C. F. Marshall, M.I.C.E., Superintending Civil Engineer, Admiralty; Mr. John Morgan, M.I.E.E., chief regional engineer, Welsh and Border Counties Region, General Post Office; Mr. L. G. Northcroft, M.I.Mech.E., Regional Fuel Efficiency Committee, South-Western Region; Mr. P. H. Pettiford, M.B.E., A.M.I.Mech.E., deputy superintendent, Royal Mint; Mr. W. W. S. Robertson, M.I.Mech.E vice-chairman, Eastern Regional Board for Industry; Mr. G. M. Sisson, general manager, Optical Works, Sir Howard Grubb, Parsons and Company, Limited; Mr. G. M. C. Taylor, M.C., M.I.C.E., senior partner, John Taylor and Sons, consulting engineers; Mr. Frank Whyte, M.C., B.Sc., M.I.C.E., engineer and manager, Caledonian Canal; Lt.-Colonel E. Woodhouse, Inspecting Officer of Railways, Ministry of Transport; and Mr. Joseph Wright, F.R.Ae.S., director and general manager, Dunlop Rim and Wheel Company, Limited. The distinction of M.B.E. has been conferred upon Mr. James Anderson, joint managing director, Anderson, Boyes and Company, Limited; Mr. J. Arnold, works production manager, Dubilier Condenser Co., Ltd.; Mr. William Buchanan, chief production engineer, G. and J. Weir, Limited, Glasgow; Mr. A. W. Burge, machine-shop and tool-room superintendent, De Havilland Engine Company, Limited; Mr. D. C. Evans, principal scientific officer, Ministry of Supply; Mr. C. A. Gutsell, chief designer, Elliott Brothers (London), Limited; Mr. J. R. Henderson, general manager, Weldall & Assembly Ltd.; Mr. J. R. Hickish, M.I.N.A., Naval Constructor, Admiralty; Mr. W. Hunter, chief draughtsman, Northern Manufacturing Co., Ltd.; Mr. P. A. Job, director, Llanelly Steel Co. (1907), Ltd.; Mr. Alexander Kilgour, A.M.I.E.E., assistant telephone manager, Glasgow, General Post Office; Mr. Lionel Lynes, M.I.Mech.E., technical assistant for carriages and wagons, Railway Executive Southern Region; Mr. G. G. Marsland, M.I.C.E., local fuel overseer, Cheltenham; Mr. T. R. Martin, general manager, metal Box Co., Ltd.; Mr. E. L. Morland, M.I.E.E., Merseyside and N. Wales Electricity Board, British Electricity Authority; Mr. David Mudie, M.I.Mech.E., J.P., chairman, Lord Robert's Memorial Workshops, Dundee; Mr. William Ross, M.I.E.E., senior executive officer, South-West Scotland Division, British Electricity Authority; Mr. H. R. Rowbotham, joint managing director, Joshua Bigwood & Son, Ltd.; Mr. W. R. Shann, A.M.I.Mech.E., Engineering Inspector, Crown Agents for the Colonies; Mr. Ernest Waldron, A.M.I.E.E., electrical engineer, Admiralty; and Mr. Ernest Wolsey, A.M.I.Mech.E., Engineer, Grade I, Ministry of Supply.

## PERSONAL

**Mr. E. Bruce Ball, M.I.Mech.E.,** has been elected chairman, and **Mr. J. M. Storey,** vice-chairman, of the British Valve Manufacturers' Association. **Mr. C. H. Armstrong, Mr. R. A. Blakeborough, Mr. O. E. Dickinson** and **Mr. N. J. Hopwood** have been elected to the executive committee of the Association.

**Mr. F. Brown** and **Mr. T. M. Elliott** have been appointed special directors of Arthur Lee & Sons Ltd., Sheffield.

**Dr. W. F. Chubb, B.Sc., F.R.I.C., F.I.M.,** has joined the staff of **Mr. R. H. Harry Stanger, A.M.I.C.E., A.M.I.Mech.E.,** Testing works and chemical laboratories, Broadway House, 24 Tothill Street, London, S.W.1, as chief metallurgist.

**Mr. R. H. Collingham, M.Inst.C.E., M.I.Mech.E.,** has relinquished his position as chief turbine engineer for The British Thomson-Houston Co., Ltd., Rugby, but will continue his association with B.T.H., as consultant on technical design problems. **Mr. K. R. Hopkirk, M.A.,** assumed responsibility for turbine and gearing design and engineering activities of the Company. **Mr. P. D. Morris, A.M.I.Mech.E.,** has been appointed chief turbine engineer and **Mr. S. A. Couling, A.M.Inst.C.E., A.M.I.Mech.E.,** chief gear engineer.

**Rear-Admiral (E.) W. G. Cowland, M.I.Mech.E.,** has been appointed Deputy Engineer-in-Chief of the Fleet.

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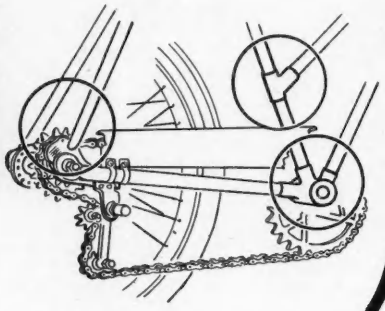

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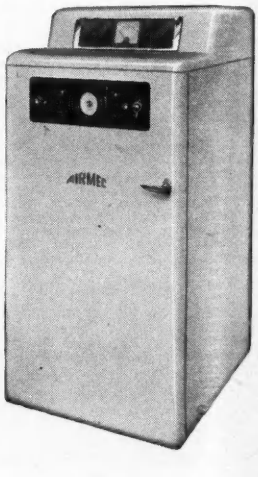


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**Mr. C. B. M. Dale, B.Sc., M.I.Mech.E.,** has relinquished his position as chief engineer of the engine division of the Brush Electrical Engineering Co., Ltd., to take up the appointment of head of the design department of W. J. Fraser & Co., Ltd., Dagenham, Essex.

**Mr. Samuel Dickinson, O.B.E., A.M.I.Mech.E.,** has been appointed head of the Department of Engineering and Mining at the Technical College, Bell Street, Wakefield.

**Sir A. H. Roy Fedden, M.B.E., D.Sc., F.R.Ae.S., M.I.Mech.E.,** has been appointed to the board of Leyland Motors Ltd., as director of research and development. At the firm's London Office, he will be dealing with long-term research and development. **Mr. H. S. Ford,** manager of the Chorley branch works, has been appointed assistant general manager of the Company.

**Mr. A. J. Filler** has been appointed Director-General of Building Materials, Ministry of Works, Lambeth Bridge House, London, S.E.1.

**Mr. J. Gordon Frater** has been appointed manager of the London sales organisation of Brookhirst Switchgear Ltd., at 54 Victoria Street, S.W.1. **Mr. C. A. Phillips** has been appointed manager of the Birmingham office of the Company at 95 Colmore Row, Birmingham 3.

**Mr. L. G. Frise, B.Sc., F.R.Ae.S.,** has been appointed chief engineer of Percival Aircraft Ltd., Luton Airport, Luton, Bedfordshire.

**Mr. Harry Green** has been appointed chief engineer (designs) of the Metropolitan-Cammell Carriage & Wagon Co., Ltd., Saltley, Birmingham.

**Mr. J. W. Grose** and **Mr. G. H. Black** have been appointed directors of Kestner Evaporator & Engineering Co., Ltd., 5 Grosvenor Gardens, London, S.W.1.

**Mr. D. B. Jephcott, A.M.I.Mech.E.,** has been appointed senior assistant in engineering at the Technical College, Wednesbury.

**Mr. F. A. Kerry, B.Sc., A.F.R.Ae.S.,** has been appointed chief designer of Saunders-Roe Ltd., 49 Parliament Street, London, S.W.1.

**Mr. A. H. Langford, M.I.P.E.,** director and general manager of Heenan & Froude Ltd., Worcester, has been appointed a director of three of the firm's subsidiary companies, namely, Court Works Ltd., Madeley, Salop ; Industrial Waste Eliminators Ltd., London ; and the New Destructor Co., Ltd., Worcester.

**Mr. I. C. de Martonfalvy** has been appointed chief engineer in the industrial lubricants department of the Vigzol Oil Co., Ltd., 111 Park Street, London, W.1.

**Mr. H. E. Newnham, R.C.N.C., A.M.I.N.A.,** chief constructor, Admiralty, has been appointed Warship Production Superintendent, N. Ireland Area and Principal (Ship) Overseer, Harland & Wolff Ltd., Belfast.

**Mr. Alexander Penman** has been appointed works manager in the carriage and wagon engineer's department of British Railways, Germiston Works, Glasgow.

**Mr. E. M. Saunders** has been appointed sales manager of the Industrial Division of Electro Hydraulics Ltd., Warrington.

**Captain (E.) W. K. Weston, O.B.E., R.N.,** has been appointed for duty with the Chief Naval Representative, Ministry of Supply, at the Naval Marine Wing of the National Gas Turbine Establishment, which is to carry out development work on marine gas-turbine plants for Naval purposes. **Captain C. W. Byas, R.N.,** has been appointed Director of Naval Aircraft Development and Production, and Deputy Chief Naval Representative (Air), Ministry of Supply.

## B.T.H. SUMMER SCHOOL IN ELECTRICAL ENGINEERING, 1948

*Proceedings.* The British Thomson-Houston Co., Ltd., Rugby, published in book form the lectures which were delivered during the Summer School, held at the invitation of the Company, to a distinguished audience of Professors of Electrical Engineering. The book can be considered as a textbook of advanced electrical engineering of great technical and educational value.

The papers published in this valuable volume of Proceedings are introduced by a Foreword by Mr. E. H. Ball, managing director of the B.T.H. Company, an Introduction by Mr. L. J. Davies, director of Research and Education and the opening address by Sir John Cockcroft on The Needs of the Laboratories and Industry of the Country for Scientific and Technical Staff.

The following is a list of the Lectures :—

Mechanical Aspects of Electrical Engineering, by W. J. Carfrae ;
Magnetic Sheet Steel, by D. Edmundson ;
Airgap Flux Distribution, by B. Adkins ;
Applications of the Rotating Harmonic Wave Theory, by Dr. J. H. Walker ;
Theory of Flux Penetration into Solid Iron and its Practical Application, by W. J. Gibbs ;
Recent Developments in Armature Windings of Polyphase Commutator Motors, by A. C. Lane ;
Some Modern Developments in Electrical Insulation, by R. Snadow ;
Vibration and Noise Problems in Engineering, by C. A. Mason ;
High Voltage Transformer Insulation Design, by H. L. Thomas ;
Power System Analysis, by Dr. J. R. Mortlock ;
The Theory and Practice of Protective Gear Circuits, by J. G. Wellings ;
Principles of Design and Applications of Switchgear, by J. M. Goodall ;
Electronic Control of Motors, by Dr. A. L. Whiteley ;
Practical Examples in Electric Wave Transmission, by Dr. L. W. Brown ;
High Power Pulse Generation, by W. S. Melville ;
High Speed Electronic Analogue Computing, by D. J. Mynall ;
Multi-Resonator Magnetrons, by L. Rushforth ;
Theory of Communication, by Dr. D. Gabor ;
Bridge Measurements at Large kVA, by Dr. K. J. R. Wilkinson ;
Thyatrons and Ignitrons—Problems of Arc Control, by H. de B. Knight ;
A Recent Betatron Development, by Dr. K. J. R. Wilkinson ;
Leak Detection, by Dr. C. J. Milner ;
A Crystal Frequency Convertor with Low Conversion Loss in 10,000 m/c Band, by J. H. Evans ;
Some New Circuits Applied to Modulator Practice, by Dr. K. J. R. Wilkinson ;
Problems in the Development of Electric Discharge and Fluorescent Lamps, by H. R. Ruff ;
The B.T.H. Educational System, by C. Grad.

Copies of the Proceedings will be sent to executive engineers in Government and Industrial Undertakings and to Professors of Electrical Engineering and Engineers holding senior academic appointments in educational establishments. A few copies are also available to other highly placed officials.

## MINISTRY OF SUPPLY PRESS TOUR OF EAST ANGLIA

Although renowned mainly for its agricultural produce, East Anglia is also the seat of a substantial engineering industry. Owing to the limited time available for their tour, extending from the 13th to 15th December, 1948, correspondents of the technical press only had an opportunity to visit a handful of the firms located in this area, but they saw enough to be convinced that East Anglia does an excellent job, not only in producing and servicing the agricultural machinery so urgently needed on its fertile soil, but also in manufacturing general engineering goods for both the home and export markets.

The first firm on the itinerary was Cocksedge and Co., Ltd., Ipswich, specialists in the manufacture and erection of complete diffusion batteries for the extraction of sugar in solution from the shredded beet. Another section of their plant, the Constructional Department, produces a great variety of welded and rivetted structures, bridges, tanks, crane runways, etc.

Ransomes, Sims and Jefferies Ltd., also in Ipswich, are reputed to be the world's oldest engineering company, having been founded in 1789 to develop agricultural machinery. Perhaps their most interesting product is a small mass-produced tractor for horticultural

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purposes. An output of 50 tractors per week is aimed at, but this target has not yet been achieved. The variety of their products is most impressive. In addition to all kinds of agricultural machinery, their products range from trolley buses and electric trucks to small milling machines.

The next firm to be visited was Cranes (Dereham) Ltd., where trailers of capacities ranging from 3 to 150 tons are being produced. Although employing only 110 people, this small plant has produced the largest vehicles in the world and the firm is now engaged in the design of a new 150 ton trailer with hydraulic suspension, a model of which was available for inspection.

Another Dereham firm, Hobbies Ltd., is universally known for its fret-saws and tools for the amateur craftsman. This firm, too, is spreading out in other directions and many contracts are being undertaken for Government departments, especially the aircraft side.

The Cooper Roller Bearings Co., Ltd., King's Lynn, was founded in 1895 to develop a steam-driven digger. The first machine proved to be inefficient and excessive friction was thought to be the cause. It occurred to the founder of the firm, the late Mr. T. Cooper, that by boring with the idler pinion and filling the space with steel rollers, this problem could be overcome, and indeed this remedy proved to be the first successful design of a roller bearing. At the present time the firm is famous for its " Split " roller bearings. The Managing Director, Mr. C. A. Ablett, who recently returned from South America, addressed the correspondents and exhorted British manufacturers to advertise more extensively there, to print their catalogues in Spanish and Portuguese, and to boost more the quality of British products. He said that British catalogues were scarce in South America, which represented a great potential market for British machinery. The South Americans, however, had the impression that the United States was the only place to buy machinery, because North America flooded them with literature and advertising about its products in Spanish and Portuguese, the languages that sell goods best in South America.

The last firm to be visited was Alfred Dodman and Co., Ltd., manufacturers of boilers, beer vats and many other products, among them high-speed steam generating sets, mainly for export to Scandinavia. Material and coal difficulties are impeding the expansion of this firm.

## BUSINESS NOTES

**British Insulated Callender's Construction Co., Ltd.,** Bical House, Bloomsbury Street, London, W.C.1, commenced operation on January 1st. The board comprises Mr. P. V. Hunter, C.B.E., M.I.E.E., Chairman ; Sir Johnstone Wright, M.I.C.E., M.I.E.E., Deputy Chairman ; and Messrs. H. F. Akehurst, B.Sc., A.M.I.E.E., C. S. Bell, W. C. Handley, M.I.E.E., W. G. Hendrey, M.I.E.E., and H. J. Stone, M.C., M.I.E.E. The executive directors are Mr. C. O. Boyse, B.Sc., M.I.E.E., general manager ; Mr. O. J. Crompton, M.Eng., M.I.E.E., manager, traction contracts ; Mr. C. H. Frankland, M.I.E.E., manager, cable and overhead line contracts ; and Messrs. E. de B. Hart, M.I.E.E., and F. B. Kitchin, joint managers, development and special projects. Mr. W. E. Tipton is secretary of the Company.

**English Numbering Machines Ltd.** have completed their move to Queensway, Enfield, Middx. The new telephone number is HOWard 2611-5.

**Kresta Foundry & Engineering Co., Ltd.,** manufacturers of hot brass pressings and non-ferrous forgings, of 33 Everton Brow, Liverpool 3, have formally opened their new modern works at Woodend Avenue, Speke, Liverpool 19, on the 6th January.

**Co-operation in Connection with Gas Turbine Developments and Application.** An agreement has been concluded between Messrs. W. H. Allen, Sons & Company, Limited, of Bedford, and the Bristol Aeroplane Company, Limited, for collaboration on the application of gas turbines for marine auxiliary purposes and in the industrial field, and for the exchange of information on aeronautical and industrial turbine problems.

**Marconi Measuring Instrument Exhibition.** Marconi Instruments Ltd. are showing a wide range of Industrial Measuring Instruments at their London Showroom 109, Eaton Square, London, S.W.1. The Exhibition will be open for two weeks, March 14th to 25th (Mon. to Fri.) and all interested in electronics and process control are cordially invited to apply for admission tickets.

Among the exhibits will be the latest pH Meters and Moisture Meters, a range of Industrial Recorders and a number of general-purpose instruments for widely differing applications. There will be many working models.

Tickets may be obtained either from Marconi Instruments Ltd., St. Albans, Herts., or the London Showroom.

## INDUSTRIAL FINISHES EXHIBITION

An Industrial Finishes Exhibition, the first of its kind to be presented in Great Britain, will be held at Earls Court, London, from August 31st to September 13, 1949. The scope of the Exhibition will embrace Non-Metallic Coatings, Natural Finishes and Metallic Coatings and Ancillary Processes and Equipment.

A great variety of finishes and processes will be exhibited and visitors will be able to compare their qualities for different applications. Some of the more recent advances Britain has made in the sphere of industrial finishes will also be demonstrated.

The practical work of organising the Exhibition has been undertaken by the Armstrong-Woridale Organisation, 26 Old Brompton Road, London, S.W.7.



# Classified Advertisements.

*The Rate for all classified advertisements is 6d. per word ; in bold print 9d. per word ; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.*

## WORK WANTED

**BOTTLENECK IN CASTINGS ?** We can probably help you. Small part of our capacity available at the moment for precision ALUMINIUM gravity die castings, ZINC pressure die castings. High standard of workmanship ; established 1828.—Write or 'phone **Thos. Ashworth & Co., Ltd.** (Dept. A/15/1), Vulcan Works, Burnley, Lancs. (Burnley 3505).

## MACHINERY, ETC., FOR SALE

**AN AUCTION SALE OF LIGHT ENGINEERING EQUIPMENT, MACHINERY AND STOCK IN TRADE** at Portslade, Sussex, on January 28th, 1949, includes :— Standard Twin Head Buffing Machine, POWER AND FOOT GUILLOTINES, Denbigh and Edwards Fly Presses Nos. 3, 4, 5 and 6, Portable Compressor, PLAIN SURFACE GRINDER WITH MICRO-ADJUSTMENT, Black & Decker 6 in. Junior Bench Grinder, " Atlas " 5 in. Lathe, PLAIN HORIZONTAL MILLING MACHINE (16 in. Bed), " Excel " sawing and Filing Machine, " EXACTA " CAPSTAN LATHE 5/8 MAX, 18 and 19 S.W.G. SHEET METAL PLATES, INSTRUMENTS—VERNIERS, MICROMETERS, &c., Complete set of tools for production of Hydraulic Windscreen Wiper for Aircraft, Loud Speaker Housings, 70,000 Switch Escutcheon Plates. Toggle Switches.

View days : Monday, Tuesday, 24th and 25th January, 9.30 a.m. to 3.30 p.m.

Catalogues from Office of Auctioneer: **H. W. M. Watts, F.A.I.,** 154 Church Road, Hove, Sussex. Telephone : Hove 6530.

**LATHES Urgently Wanted :** Modern motorised machines. Also Universal Milling machines, Radial Drills, Surface Grinders 24 in. × 6 in., Ward 2A and 3A Capstans. Combined Wood Planers Thicknessers. Good prices for Modern tools.—**Victa Engineering Company, Maidenhead.** 'Phone : 50.

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